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SWINGLE'S PRACTICAL HAND-BOOK FOR MILLWRIGHTS

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PRACTICAL HAND BOOK FOR MILLWRIGHTS

DESCRIBING THE PRACTICAL PLANNING AND ARRANGEMENT OF MILL BUILDINGS; STRENGTH OF MATERIALS; MODULUS OF ELASTICITY; MOMENT OF INERTIA; SHAFTING; JOURNAL BEARINGS; COUPLINGS; HORSEPOWER TO BE TRANSMITTED BY SHAFTING; ALIGNMENT AND LEVELING OF SHAFTING; PULLEY LOCATION; KEY FITTING; SET SCREWS; BELTS AND BELTING; WIRE ROPE TRANSMISSION; GEAR WHEELS; ELECTRICAL INSTALLATION AND ALL SUBJECTS DIRECTLY CONNECTED WITH THE INSTALLATION OF MILL MACHINERY; ETC.; ETC.

BY

CALVIN F. SWINGLE, M. E.

Author of "Encyclopedia of Engineering," "Examination Questions and Answers for Marine and Stationary Engineers," "Modern Locomotive Engineering" and "Modern Steam Boilers."

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Handbook for Millwrights

INTRODUCTION

A millwright, according to Webster, is "one who makes a business of planning and building mills or mill machinery." Therefore in order to become a successful millwright, a man must also be in a certain sense an architect. He should know how to plan, and erect a mill building from foundation to roof truss, in such a manner as will be best adapted to the type of machinery he will be called upon to install. He must be able to calculate strains, the strength of materials, and the resultants of forces. He should be able to make and to read drawings, not only of machinery, but of foundations also, and he should possess technical knowledge sufficient to enable him to make correct calculations regarding the dimensions of foundations, not alone for the buildings, but also for the boilers, engines, and other heavy machines.

He should also be a man full of resources and determination, ready to meet and cope with any emergency or problem which may arise. He should possess the ability to handle men (both skilled and unskilled workmen) in such a manner as to get the best results from their labor. He is frequently called upon to purchase machinery and supplies; therefore he should be a good business man and accountant. The increased, and increasing use of electricity as an agent for the transmission of power in mills and factories has added another to the list of qualifications of the modern millwright, viz.: a good working knowledge of electric machines, such as generators, transformers, motors, rotary converters, electric lights, and the various other electric apparatus with which the modern up-to-date manufacturing establishment is equipped. A complete knowledge of the framing of timber, and the manipulation of structural steel, together with the construction of concrete piers should also be the aim of every man who aspires to become a successful millwright, and he should bear in mind that improvements and changes are continually being made in methods and materials pertaining to his calling. Therefore, he should be a man of study, and should endeavor to make himself thoroughly familiar with all new and improved apparatus, provided it possesses merit. By thus devoting a reasonable portion of his time to reading and study, he will be able to keep step with the march of progress, and always be at the front.

LOCATION OF BUILDINGS.

The location of factory buildings is usually decided upon by the owner, or superintendent, but the millwright is frequently called upon for advice in such cases, and if he is a competent man he will be able to make many valuable suggestions regarding details relating to the best methods of receiving the raw material, in the factory, and the disposal of the finished product. A good rule to follow in designing a manufacturing plant is to see that the course of the material under operation shall, from the time of its reception at the factory, until its exit as a finished product, be in one general direction, viz.; forward in as near a straight line as possible. It costs money to handle material and any deviation from the above rule will result in increased cost of production. For instance, suppose the building has been so planned, and the machines so located that the material must in the regular course of manufacture be transported a second, or third time to the same floor in order to be worked upon, the added cost of its conveyance, especially if by elevator to a second, or third story, will greatly affect the efficiency of the plant. It is possible to so locate the various machines in the mill in such a manner that the material in its progress need never pass a given point the second time, and this fact should be borne in mind by the millwright in drawing up his plans. Another very pronounced cause of inconvenience and loss of time in doing the work, is the lack of working space around the machines. In the laying out of the different floors, provision should be made for space sufficient around each machine to allow the man in charge of it to operate it without interfering in any way with those next to him.

FOUNDATIONS.

The first thing to be considered in the erection of any structure, is of course the foundation.

The object of foundations is to prevent inequality of settlement and distribute the weight of the structure equally over the substratum.

The bases of structures are invariably made wider than the superincumbent mass, to increase the stability and to counteract all the damaging forces that tend to cause failure.

Damaging Forces.—The principal causes of failure are those which induce settlement, such as inequalities of earth resistance; the compressibility of mortar joints; lateral escape of soft soil, sliding of the substratum on sloping ground; the withdrawal of water; distributed lateral pressures, causing overturn, such as wind pressure, and thrust of barrel vaulting or of an untied couple raftered roof; concentrated lateral pressure which induces settlement and overturn, such as the thrust of framed floors, trussed roofs and groined vaults subjecting small areas of support to great pressures.

In the case of mill buildings in which heavy machinery is to be operated, there is to be added to the foregoing list of damaging forces, the jar and strain of the moving machinery.

Inequality of Settlement.—Inequality of settlement in buildings takes place from two causes: (1) the compressibility of the mortar joints; (2) the compressibility of the soil.

An allowance of 1 in. in 24 ft. of brickwork in lime mortar is often provided for settlement, as in the example of the extremities of bridging joists of floors, at one end being supported by a brick wall and the other extremities by iron columns, etc.

Nearly all soils, with the exception of solid rock and gravel, are compressible under pressures often attained in buildings. It is therefore impossible, where large buildings are erected on other soils, to avoid settlement; and the fact of any building settling is of no great import, provided the settlement be uniform and of no great depth, and the relative position of the parts of the structure unaltered. But where the resistance of the soil of every part of the site is not uniform, there is a risk of the above defect occurring, and special precautions must be taken to distribute the pressure to suit the varying strengths of the substratum.

Lateral Escape.—Heavy structures erected upon soft soils, such as running sands and peat, squeeze out from beneath the foundation, unless means are taken to confine the soil to the required area; this is usually accomplished by sheet piling, as described later.

Sliding.—This is a defect usually occurring where the building is erected on the slope of a hill, and the strata inclined, being depressed in the direction and towards the bottom of the slope. The weight of the building is liable to cause the strata to become detached and slide. This is prevented in two ways: (1) by driving piles at intervals to a considerable depth, thus connecting the strata; this method is often objectionable, tending, as it does, to shake and disturb the soil; (2) by building a retaining wall; this is the better method, as it not only supports, but also protects the strata from the effects of the atmosphere, which in soils easily affected by the latter is a desideratum.

Withdrawal of Water from Foundation Earth.—Edifices built on damp soil, such as a sand overlying a clay, have their stability endangered, should the water be drained away after the building has been erected, as it will cause the foundation earth to occupy a less volume and in the sinking will tend to fracture or overturn the walls; therefore the depth of the concrete foundation must be arranged below any probable adjacent cutting.

Distributed Overturning Pressures.—Distributed forces acting upon the upper level of walls, such as the continuous pressure of barrel vaulting, and the spreading tendencies of untied couple raftered roofs, and also the distributed pressures on wall faces, such as wind pressure, tend to cause failure in two ways: (1) by overturning, the minimum resistance being generally at the change of section, usually at the ground level; (2) by subjecting the leeward edge of the wall to the pressure sufficient to crush the material, or by throwing the weight on a small area of the substratum, forcing it from its original position and causing a settlement.

The safe load which may be placed upon any foundation depends upon the carrying power of the soil and the area of the foundation footing. It is not the purpose of the author to enter into a discussion of the carrying power of the various kinds of soil met with, therefore nothing will be given in that direction save the following table which will serve as a guide to the millwright when he must determine whether or not the plans furnished call for an excessively heavy foundation, or whether the mass of material shown underneath some heavy machine is insufficient for the particular soil met with during the operations at hand.

TABLE 1.—BEARING POWER OF SOILS.
By Prof. Ira C. Baker, University of Illinois.

	Tons. sq. Ft	
Kind of Material.	Min.	Max.
Rock—the hardest—in thick layers, in native bed	200	..
Rock equal to best ashlar masonry	25	30
Rock equal to best brick masonry	15	20
Rock equal to poor brick masonry	5	10
Clay on thick beds (always dry)	4	6
Clay on thick beds (moderately dry)	2	4
Clay, soft	1	2
Gravel and coarse sand, well cemented	8	10
Sand, compact and well cemented	4	6
Sand, clean, dry	2	4
Quicksand, alluvial soils, etc.	0.5	1

Concentrated Lateral Pressure.—The thrust caused by united principals, as groined faults or other forces acting at a point, or along vertical lines on the wall, are often resisted by buttresses.

Atmospheric Action.—Many otherwise thoroughly reliable soils are practically reduced to the condition of mud if exposed to the effects of the atmosphere or to rain water. The variation in temperature at the different seasons also causes the ground to expand and contract considerably.

Where foundations are constructed in such soils, they must be taken sufficiently deep to be beyond the effects of the atmosphere, that is, below the line of saturation.

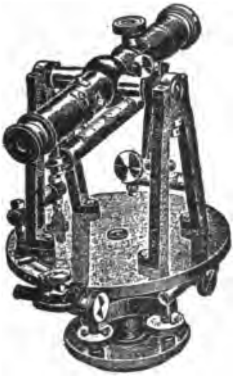


FIG. 1.
BUILDER'S TRANSIT.

The line of saturation in the section of any part of the earth's crust represents the depth to which the soil at that part is saturated by the absorption of rain water and affected by atmospheric changes.

Excavations.—Before commencing any constructional work in connection with a building, it is necessary as the first operation to carefully take the levels of the site, in order first to arrive at an estimate of the amount of earthwork to be done; and secondly, to determine the design of the basement story, this latter often being materially affected if the differences in level of the various parts of the site are great. The next operation is to level the ground. This in most instances consists in excavating and removing parts of the site, and in depositing earth in other parts to form embankments or to fill up hollow places. In order to conduct these operations in the most economical manner the levels must in all instances be taken and plotted with the greatest accuracy. This can only be efficiently done on areas of any magnitude by means of the surveyor's level, the method of employing which will be described later. All leveling operations for ordinary constructional work may be carried out by referring them to the principles laid down for performing the three following operations:

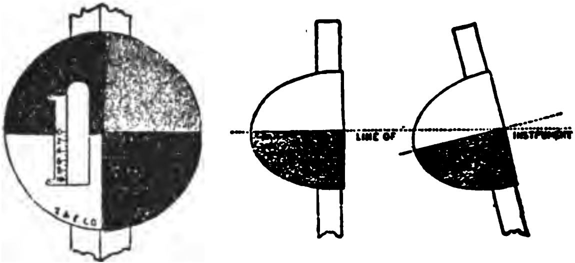


FIG. 2.
ROLLING ANGLE TARGET.

1. Taking levels of site:
2. Leveling the bottoms of trenches for drains or foundations.
3. Embanking for roads, or leveling of depressions.

Instruments.—The instruments required to determine the levels of the site are: first, the surveyor's level; second, the measuring staff; third, ranging poles and chain or tape.

Methods of Leveling.—Taking the levels of a site may be carried out in one of three ways: First, by taking a number of section lines across the site; secondly, by erecting the level in a commanding position and taking the relative heights of the salient points and noting them on plan (this method is only applicable for small sites); thirdly, by contours.

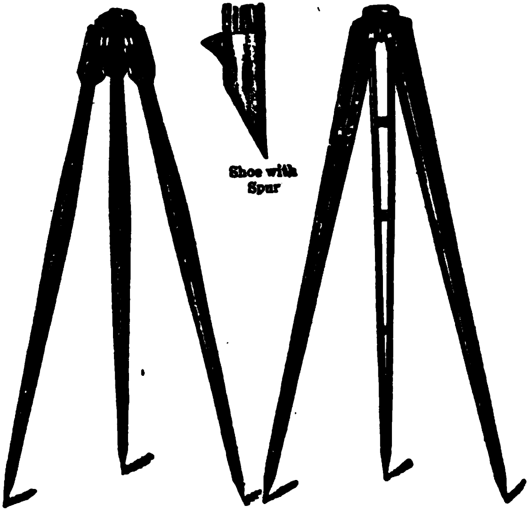


FIG. 3.
TRIPODS FOR LEVELS AND TRANSITS.

In all three methods it is necessary to have a datum level to commence from, and to which all other levels can be referred. A line on some permanent structure in the immediate vicinity is usually taken, or if such does not exist, a stout stake is driven in the ground in a position away from the work where it is not likely to be disturbed.

First Method.—A number of sections are ranged across the site, each line being numbered or lettered; the level is then set up on, or in close proximity to the first line and the datum; the measuring staff is then held by an assistant on the datum point and then on the extremity of the line, the relative heights of the two points being recorded in a field book kept for that purpose. A number of points on the line are then taken, and the measuring staff is held over them and their relative heights are recorded, and their distances from the beginning of the line are measured. When the bottom of the measuring staff rises above, or the top becomes depressed below the line of sight through the rise or depression of the ground, the level must be moved further along the line and the preceding

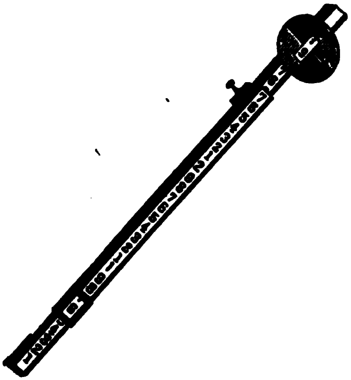


FIG. 4.

operations repeated. Fig. 5 illustrates the method. The following is a form of field book with the reading for a section entered:

FIELD LEVEL BOOK.

Back Sight.	Inter. Sight.	Fore-Sight.	Rise	Fall.	Reduced Levels.	Dis-tance.	Total Dis-tance.	Remarks.
						chains		
4.15	.....	.....	.....	....	100'.0	......	......	Bench Mark A
	4.13	.....	.02	....	100.02	1	......	1 peg
	5.01	.....	.....	.88	99.14	2	......	2
	4.86	.....	.15	....	99.29	3	......	3
	6.06	.....	.....	1.20	98.09	4	......	4
		8.02	.....	1.96	96.13	5	......	5
12.25								
	8.46	.....	3.79	....	99.92	6	......	6
	3.04	.....	5.42	....	105.34	7	......	7
12.60								
		2.15	.89	....	106.23	7.57	......	Bench Mark B
	7.19	.....	5.41	....	111.64	8.57	......	8
		2.53	4.66	....	116.30	9.57	......	9
9.37								
	5.75	.....	3.62	....	119.92	10.57	......	10
		3.94	1.81	....	121.73	11.57	11.57	Bench Mark C
			25.77	4.04	21.73			
			4.04					
			21.73					

The above shows a typical field book. The reduced level of the first point is taken as 100 ft. above a datum level; the levels are all read in feet and hundredths of a foot; the distances are taken in chains and links, but may be taken in feet and inches. The rise and fall columns should be balanced, also the first and last reading in the reduced levels; these two quantities will equal each other if the computations have been correctly made.

Second Method.—The second method is evident from the previous explanations.

Third Method.—The method of contouring is the most useful, but takes the longest time to perform it; it consists in describing upon a plan a series of level lines with a uniform vertical interval between them.

To carry out this operation it is usual to erect the instrument on the highest point of any section of the area to be contoured, and from this point to range a number of lines radiating from it, their direction being fixed by taking their bearings. The height of the instrument is then taken, and the man with the measuring staff is directed up or down each line in succession until a number of points of the required vertical interval and their distances from the initial point are determined. This method is most useful for laying out large estates where extensive works are projected, as on such a plan the problems of drainage and roads of convenient and economical gradients can easily be laid down.

When the levels of a site are known, and the building is planned, and the location of two of its leading lines are determined, to set out the remaining lines of an ordinary building becomes a simple matter, only requiring great care in the measurements of the parts. If the setting out is rendered difficult through differences of level in the paths, a theodolite would very much simplify the operations.

Boning Method of Leveling.—This operation is used for the leveling of trenches, ground work, paving, etc. There are three rods in a set; two of these are leveled at a distance of about 10 ft. apart; a third rod is then leveled at a similar distance, taking care to reverse the long level. The center rod is then removed, and the level transmitted to any point along the line by sighting or boning over the first and third rods.

Fig. 10 shows the method of using boning rods and setting a curbstone.

Trenching.—When the lines of the building have been laid down and all its salient angles pegged out,

Fig. 5.

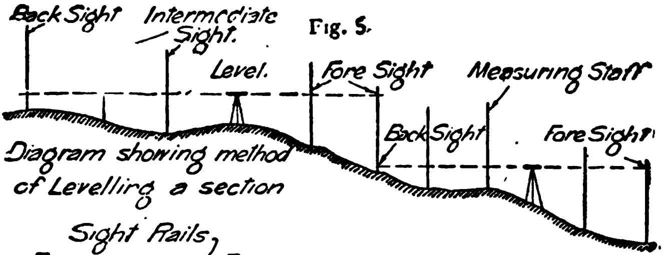


Diagram showing method of Levelling a section

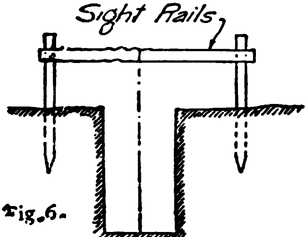


Fig. 6.

Sight Rail with Posts fixed in ground

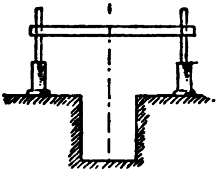
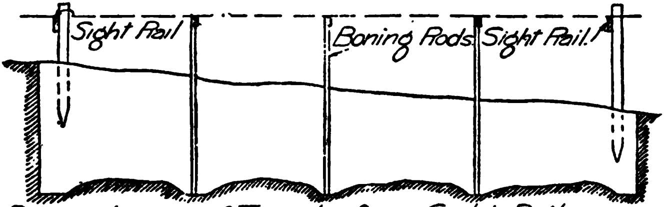


Fig. 7.

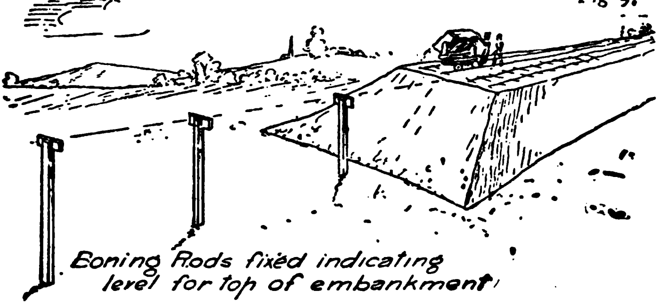
Sight Rail with Posts fixed in drain pipes

Fig. 8.



Boning bottom of Trench from Sight Rails

Fig 9.



Boning Rods fixed indicating level for top of embankment

METHOD OF LEVELING

the work of excavating the trenches commences. It is absolutely necessary that the trenches should be level along their bottoms. To ensure this, two or more sight rails (as shown in Figs. 6 and 7) are erected over the trench; it is necessary that the side posts of these should be fixed in such a position that they shall not be disturbed by any of the subsequent operations. A level line is sighted through the level and marked on the sight rails; the cross bar is then fixed on each, and a mark is made on the bars plumb over the center of the trench. The width of the trench is marked out with the line and pins (see Fig. 9), and the excavation is carried on, timbering being inserted as the earth is removed, if required, by one of the methods afterwards described. When the full depth of the trench has been nearly reached, a number of points are sunk to the exact depth by means of boning rods, the top of which is sighted between two of the sight rails, as shown in Fig. 8. The remaining parts of the trench bottom are then taken out level between the points so determined. A similar process is pursued for sinking a trench for a drain, the variation being that the sight rails have a difference in height necessary to give the required fall.

Embanking.—The method of forming an embankment is as follows: The center line of the proposed work is ranged out on the ground, and at equal intervals along the line boning rods are erected, the two extreme rods being first fixed either level or with a difference in height sufficient to give the required gradient; a rod is then erected on each of the intervals determined upon, and boned between the two extreme rods. The embankment is then commenced from one end, the earth being tipped in from carts or wagons until the tops of the boning rods are reached; sufficient earth in excess must be allowed for to compensate for compression and settlement. The width of the embankment is completed as the work is pushed forward, as shown in Fig. 9.

Timbering for Excavations.—It becomes necessary, where earth has to be excavated to any considerable depth, for foundations or other purposes, to support the sides of the cutting until the sinkings, or trenches are filled in, or other action taken to permanently support the sides. This end is attained by means of timber shores, the arrangement of which is modified and governed by several conditions, such as the nature of the soil, the size of the cutting, and the special peculiarities of the particular piece of work under consideration.

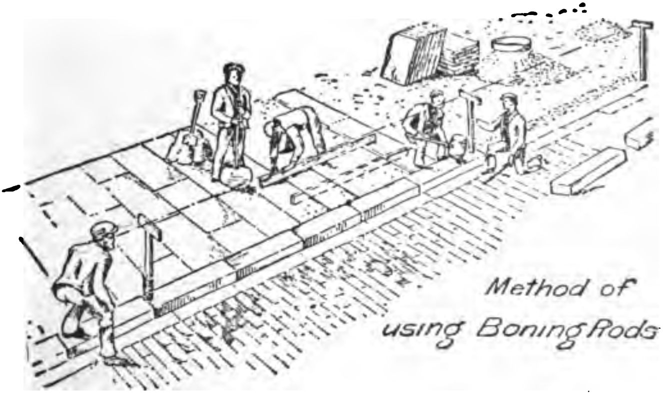


FIG. 10

There are three typical methods of strutting used for supporting the sides of narrow trenches excavated for foundations or drainage work, shown in Figs. 11 and 12.

The first, used for firm ground, consists of short upright members, termed poling boards, out of

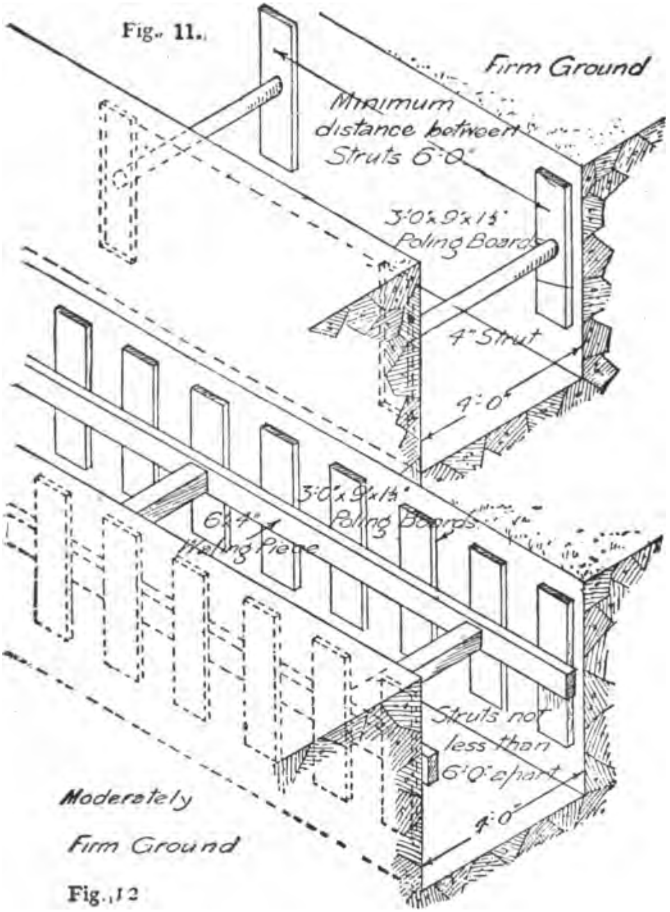

Firm Ground
Minimum distance between Struts 6'-0"
3'-0"x9"x1½" Poling Boards
4" Strut
4'-0"
3'-0"x9"x1½" Poling Boards
6"x4" Waling Piece
Struts not less than 6'-0" apart
4'-0"
Moderately Firm Ground

Fig. 11.

Fig. 12

9 x 1½ in., usually from 3 to 8 ft. long, placed in position in pairs, one board on each side of the cutting; these are kept apart by struts out of about 4 x 4 in., or

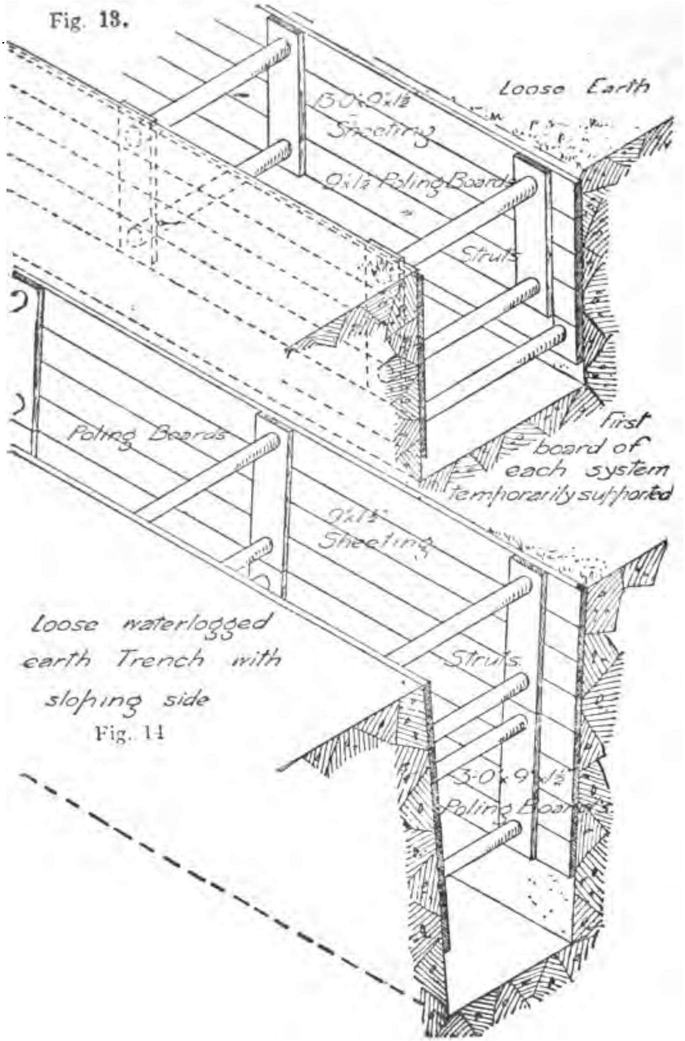
short ends of scaffold poles cut and driven tightly between the poling boards. The strutting is fixed as soon as the trench has been made sufficiently deep. The horizontal distance apart between the adjacent system of strutting varies according to the cohesive strength of the soil, but never less than 6 ft., which is just sufficient to allow a man to work in with effect.

The method shown in Fig. 12 is adopted where the earth requires to be supported at shorter intervals than 6 ft., and consists of upright poling boards and struts as before, but with the addition of a horizontal timber termed a waling piece. The process of fixing is as follows: The cutting is made, commencing at one end, and as soon as sufficient earth has been excavated a pair of poling boards and struts is inserted as in the first method; this process is repeated, fresh poling boards being fixed at distances apart varying with the nature of the earth, these distances being in some instances very short.

Horizontal members, 4 x 4 in. or upwards, are placed one on each side of cutting and strutted tightly against the poling boards. After about 12 ft. has been thus cleared, the struts which were fixed first are then knocked out; a fresh depth is commenced, and treated in a similar way.

The third method is employed where the earth is very soft, and consists in laying horizontally boards, usually 9 x 1½ in., against the sides of the excavation; the boarding laid in this manner is termed sheeting, which is supported by upright poling boards and struts, as shown in Fig. 13. The method of fixing is as follows: The earth is taken out to a depth of 9 in., and a pair of boards is inserted and strutted apart; another depth of 9 in. is then taken out, and sheeting fixed as before.

Fig. 13.



Loose waterlogged earth Trench with sloping side

Fig. 14

This process is repeated until a sufficient number of boards has been inserted, usually four; upright poling boards are then placed in position against the sheeting and strutted apart, as shown in Figs. 13 and 14; the first fixed struts are now struck and cleared away.

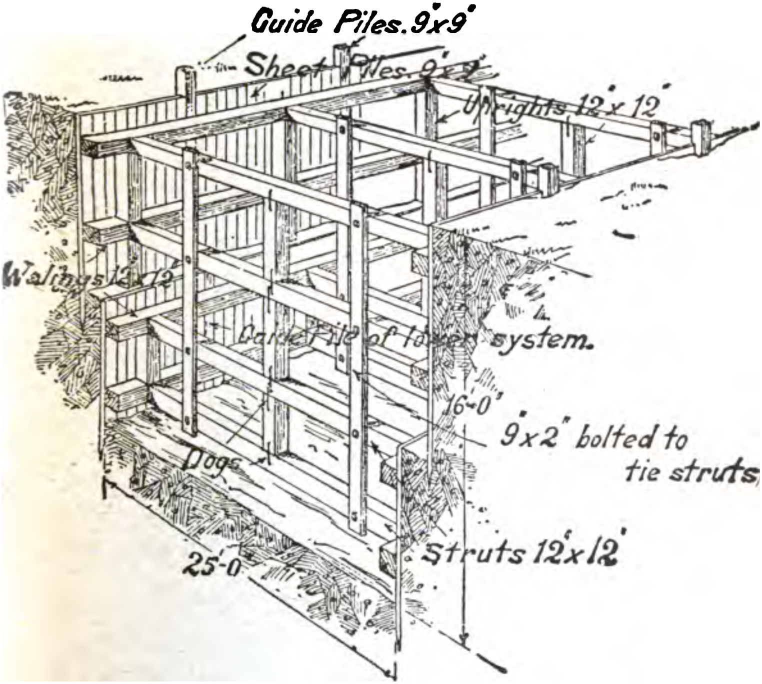


FIG. 15

The above system may be improved upon, when the depth of the cutting is not too great, by cutting the sides of the excavation to a slight batter, as shown in Fig. 14; by so doing the timbers are prevented from falling should the earth contract on becoming drained; it also facilitates the fixing of the struts.

Large Cuttings.—Continuous trenches, if made in bad ground, are generally arranged as shown in Fig. 15. At intervals guide piles are driven in, to which walings are bolted, and sheeting consisting of boards about 10 ft. long, shod with iron, termed runners, inserted between; these are driven a short distance into the ground, the earth between the two systems of piles being then taken out, and care taken not to excavate within a foot of the bottom end of the runners, which are again driven in and the process repeated. After the excavation of the first part, wales, consisting of whole timbers, are placed in position and strutted apart, the struts being also of balk timber. Long struts are supported in the direction of their length by short uprights secured to them by dogs. Uprights are also placed between the waling pieces as each fresh one is inserted.

After the ground has been excavated to the depth of the runners, a fresh system of piles and runners is driven slightly to advance of the former system, and the ground excavated as before. Cuttings are made in firm ground by excavating the earth and using ordinary sheeting, but if the cuttings are required to exceed 30 ft. in width, it is found to be more economical to adopt a system of raking shores.

The method illustrated in Figs. 16 to 18 is employed where the ground is soft and waterlogged, and is especially suitable for running sand. By this method as much of the earth is taken out as is possible without the sides of the excavation falling in, generally from 4 to 6 ft.; this is then supported by upright sheeting, waled and strutted. The excavation is continued by lining the cutting with a secondary system of runner, *i. e.,* battens 7 x 2 in., pointed at lower ends and of about 9 ft. in length. These are waled and strutted. Between each

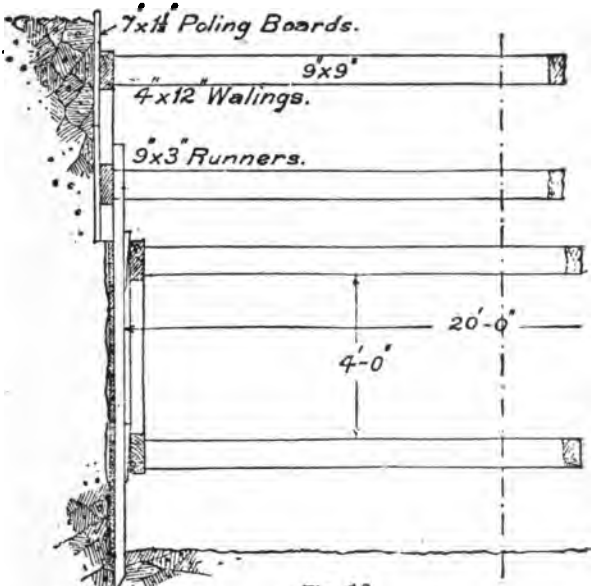

7"x1½" Poling Boards.
9"x9"
4"x12" Walings.
9"x3" Runners.
20'-0"
4'-0"

Fig 16.

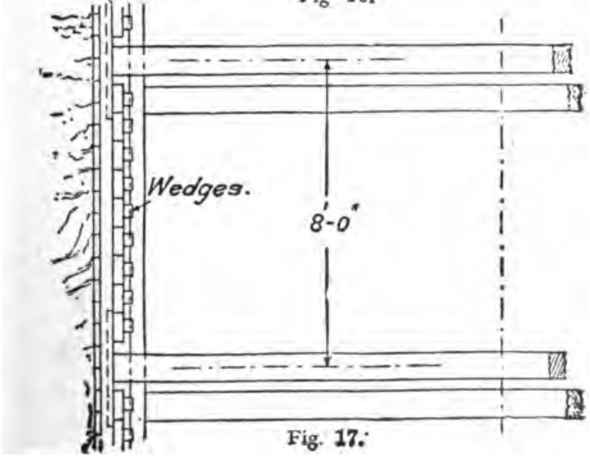

Wedges.
8'-0"

Fig. 17.

runner and waling piece a wedge is inserted. The method of proceeding with the excavation is as follows: The wedges securing one runner are loosened, the earth from the foot removed to a depth of about 12 in., the runner being dropped as the ground is removed and re-wedged. Each runner is successively treated in this manner till the whole system has been lowered the necessary amount. It is essential that the feet of these runners should be at all times kept in the ground, as, if any portion of the vertical side of the excavation be exposed, the earth is liable to ooze out and leave the back of the runners unsupported and cause the whole system to collapse.

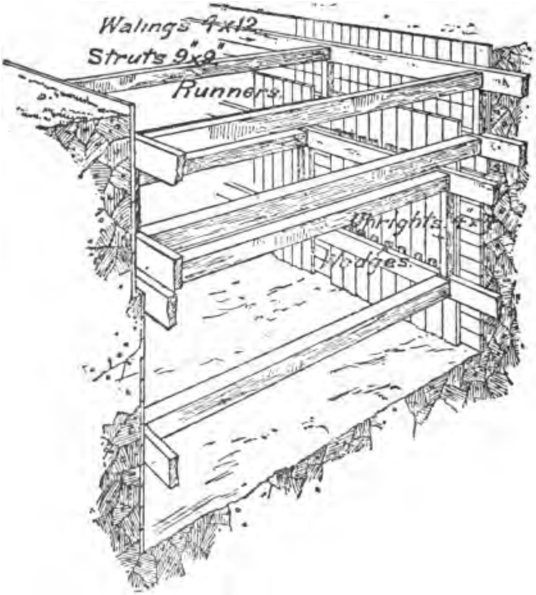


FIG. 18

Sinking Shafts.—It is often necessary to sink shafts for foundations, etc. These are made from 4 ft. square and upwards, the former being the smallest size a man can work in without difficulty.

Shafts from 4 to 9 ft. square are timbered as shown in Fig. 19.

In ordinary soils the earth is excavated to a depth of at least 3 ft., and in firm soils 6 ft. The sides of the excavation are then lined with vertical sheeting, consisting of boards 9 in. wide, 1 to 1½ in. thick, strutted apart by frames of horizontal waling timbers, a pair of which is placed in position against two opposite sides, and strutted apart by another pair driven tightly between and against the remaining sides, these being secured by cleats nailed to the fixed waling pieces. Another depth of earth is then taken out and a second system of sheeting placed in, the upper ends of which lap about 1 ft. over the lower ends of the first system of sheeting; another frame is placed in position as before, securing both systems of sheeting. Uprights are fixed in the angles between the waling pieces, and often at intermediate positions along their length. This process is repeated till the required depth is obtained.

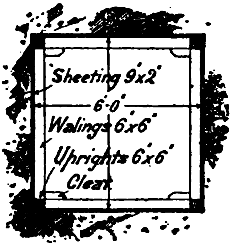


FIG. 19

The timbering requires to be supported if the depth be great, to prevent it from sliding down on the removal of the earth from its lower end. Where this has to be done, the upper end of the shaft is left projecting about 3 ft. above the ground level. The two first fixed waling timbers at the ground level are continued through the shaft, and project several feet on either side, a good bearing on the solid ground on both sides of the shaft being thus obtained, as shown in Figs. 20 and 20A.

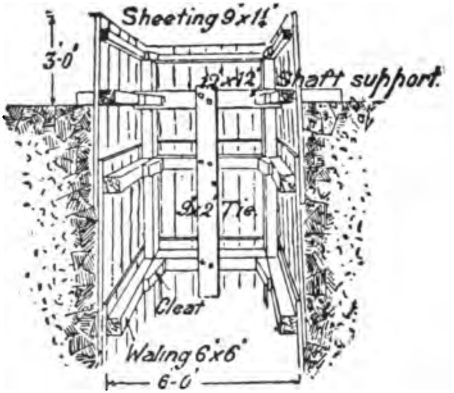


FIG. 20

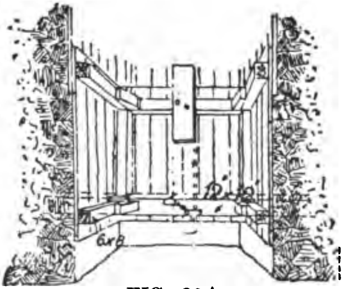


FIG. 20A

These members are usually out of square timbers; they are strutted apart as described. An upright vertical timber is notched over this, and spiked to the face of the waling timbers below, the whole being thus tied together.

These are often supplemented by similar timbers at the bottom of the shaft. These timbers are fixed in two pieces, with a scarf in the center; they project about 3 ft. into both sides of the pit. A chain is sometimes employed in addition to the timber spiked to the walings.

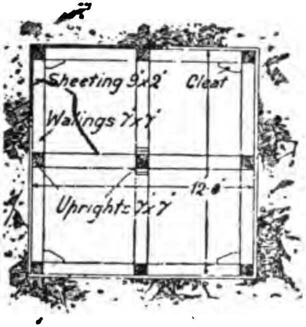


FIG. 21

Intermediate struts are required to support the horizontal walings where the size of the pit is above 9 ft. square. One system of struts is fixed between two opposite sides, being supported at their ends by cleats, as shown in Figs. 21 to 23; these being necessary to prevent the timbers falling should they become loose during the progress of the work. The struts that support the remaining sides intersect by butting, as shown in Fig. 22, against the first system, and are therefore fixed in two pieces. The struts at their intersection are supported by uprights, on the upper ends of which short ends of timber are placed, projecting beyond the sides, acting as corbels, and forming a ledge upon which the shorter struts take a bearing.

The earth is raised from the bottom of the shaft, if of a great depth, by means of hoisting tackle; but if the cutting be shallow, stages are often erected in 6 ft. heights, the earth being shoveled from one to the other till the top is reached.

Tunneling.—In building operations it is often necessary to bore a tunnel in order to construct drains, etc., the process being carried out as follows:

Tunnels are made just large enough for a man to work in, that is, from 4 to 7 ft. square. The earth is taken out in sections of about 3 ft. at a time, poling boards of the same length being then placed against the upper surface, and kept in their position by a system of strutting, consisting of a head, sill and two uprights, out of either round or square timbers. The sill is placed in position first, being partly bedded in ground to prevent lateral motion, and being bedded in its correct vertical position by boning through from the sills previously bedded; the head next, then the struts, which are cut and driven tightly between the two. The next section is then cleared out, commencing at the top, just enough being taken out there to allow of the next system of

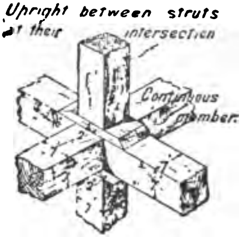


FIG. 22

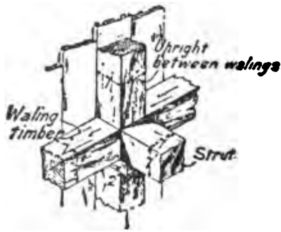


FIG. 23

poling boards being inserted, these being arranged to overlap the first system at their back end, the two being then strutted up together; this process is repeated till the tunnel is finished.

If the soil be bad and the sides liable to fall in, they must also be lined by poling boards, these being kept in their place by the uprights.

Large spikes, similar in shape to floor brads, are driven into the head and sill, with their heads left projecting so as to be easily withdrawn, to secure the struts when in position. Wood cleats are often used in place of these.

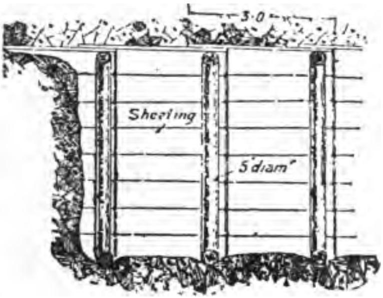


FIG. 24

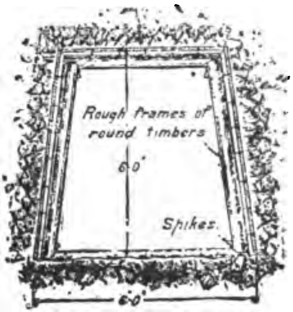


FIG. 25

These tunnels are usually made slightly tapering from the base to the head, as shown in Figs. 24 and 25.

Foundations.—The construction of foundations varies with the nature and bearing strength of the soil. The following are the ordinary soils met with in practice and the method of treating them: Rock, chalk, gravel, clay and sand.

Rock.—Foundations laid upon the solid rock are undoubtedly secure, as far as settlement is concerned; such a substratum being practically incompressible. Rocks often have fissures and defective parts, and all gaps must be filled up with concrete, any unsound parts being cut away. Rock foundations are very expensive in working, owing to the extra labor involved in cutting them; but where they occur they may be built upon direct.

Chalk.—The sites for buildings on chalk or marl soil should be drained, and precautions taken to prevent them becoming wet. Where this can be done, the structure can be built upon the chalk or marl direct, after it has been leveled: but where heavy buildings are erected, or great weights concentrated, concrete should be employed to distribute the pressure.

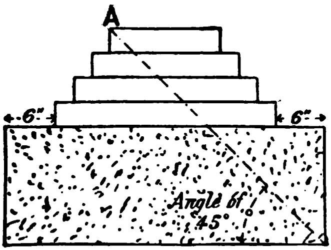


FIG. 26

Gravel.—Where lateral movement is not likely to occur, gravel is one of the best soils to build upon; it is not affected by the action of the atmosphere, and is practically incompressible.

Clay.—Clay is a good soil to build upon where the foundations are taken deep enough to be beyond the action of the atmosphere. Clay is very subject to expansion and contraction with the variations in temperature, and is therefore dangerous to build upon unless protected.

Sand.—Sand is a good material to build upon, if it can be kept dry and confined laterally; if subjected to the effects of running water it is liable to be scoured from about the foundation.

In all the above soils, with the exception of the rock, and the chalk when in a good condition, it is usual to form a bed of concrete, the area of which is proportioned to the weight to be carried and the bearing strength of the soil.

The following are cases that require special treatment: (1) Soft soils of a great depth; (2) soft soils with hard strata beneath; (3) soils not having a uniform resistance, formed of rocks which have hollows or fissures filled up with some softer material.

A good and easily applied test for the soil can be made as follows: Place a bearing one foot square upon the soil at the bottom of the excavation, and load it with stones, brick or anything having weight, until the square foot of foundation begins to squeeze its way into the soil. Then note the weight of material that was placed upon the foundation, and allow such factor of safety as is deemed necessary. The one foot square foundation may be a piece of 2 x 12 inch plank, or anything solid. In this manner the number of tons per square foot that the soil will safely carry may be estimated, and the foundations be built accordingly.

In preparing footings on which to lay bricks, care must be taken to keep the work in line and fairly level on top before the brickwork is commenced, whether the lower footings be of stone or of concrete. At this writing, concrete seems to be the popular material in use for the lowest layer of foundation, and justly so, as, when properly put in place, and the proportions of the various materials wisely assigned and mixed, the work will be as though one solid stone was laid all round the building on which the brickwork may be placed. An illustration of the proper method of laying in a concrete footing is shown in Fig. 26, and one that has been adopted in many an architect's office and many a municipal building department. Taking the wall in section and extending the concrete each side of the bottom course of footings, drop perpendicular lines as outside width of concrete, the depth being determined by an angle of 45 degrees, passing from the point A of the next work, and cutting the outside line of concrete. A cubic yard of concrete would require 27 cu. ft. of broken brick, stone or shingle, 9 cu. ft. of sand, 4½ cu. ft. or 3½ bu. of Portland cement, and 25 gal. of water. These quantities should be correctly measured, turned over together three times dry, and again several times while the water, through a hose, is being sprinkled over the mass. Broken brick or stone small enough to pass through 1½-in. mesh is preferable for the aggregate. The practice of throwing in concrete from a height, in order to consolidate the mass—which used to be considered essential, even when staking had to be erected and the stuff wheeled up to the required height at considerable expense—has now exploded. It should be brought on to the side, deposited and lightly punned or beaten down with wooden rammers, but only just sufficient to bring the moisture to the surface; if rammed too much the cement comes up with the water. If, however, it is more convenient to tip the concrete into an excavation, no sensible injury will be done to it.

The objection that, in falling, the heavier particles separate from the finer is, from the very stickiness of the mass, more theoretical than practical, and, at the most, applicable only to each separate barrow load tipped in, and not to the whole bed. Sliding it down a wooden shoot, however, should never be permitted, as the cement and small stuff cling to the sides and run down in a muddy slush; while the stones are shot out into a separate heap by themselves.

In ordinary foundations the concrete should be deposited in horizontal layers, about 2 ft. thick, and care should be taken to cover any joints in one layer by the succeeding one. as the joint between two days' work is always a weak part; moreover the past layer should be well wetted to insure a proper connection with the next.

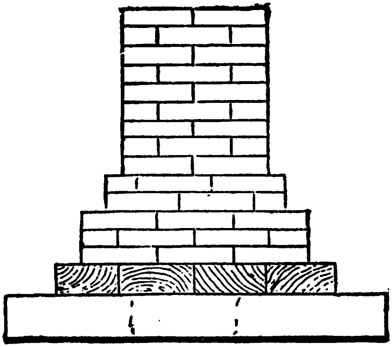


FIG. 27.

In Fig. 27 an illustration is shown of a wall and footings, the latter being of stone, not less than 6 in. thick. On the lower footing of stone is laid another course of stonework, and on this is laid the brickwork, the top of the upper stone being made level and a layer of good mortar spread over it so that the bricks

Sections of Footings and Walls in English Bond.

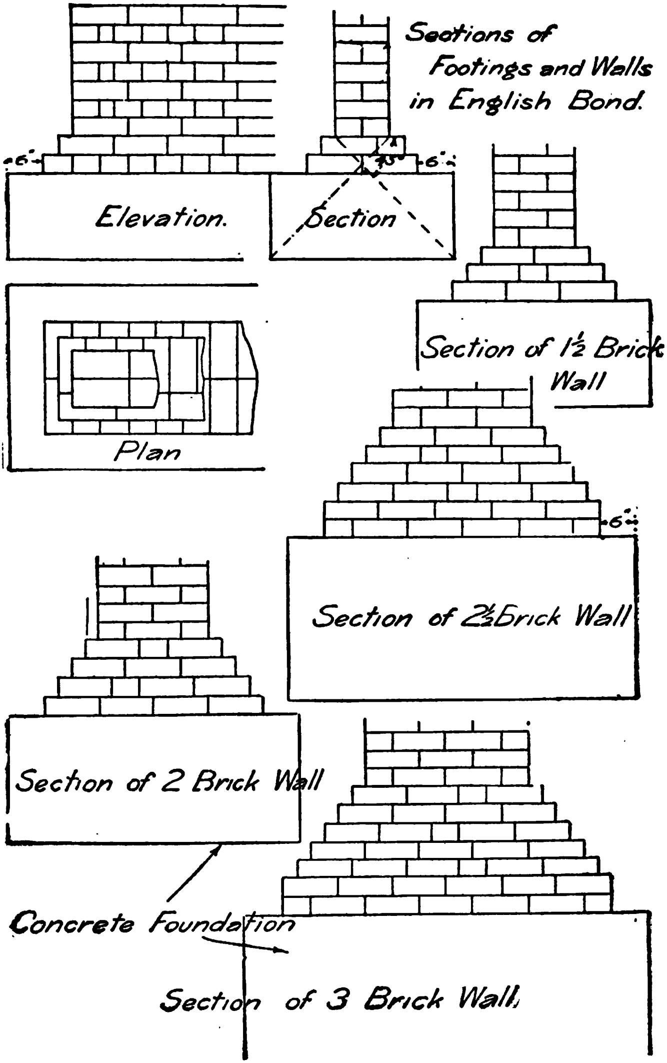


FIG. 28.

have a good bed to rest on. This layer should be cement mortar where possible, as it would help to make the whole work stronger and better.

Fig. 28 shows five sections of brick walls and footings, with the methods of arranging the bricks in the wall; there being a one, a one and a half, a two, a two and a half, and a three brick wall, showing the proportions for concrete footings. A scale in feet and inches is shown on the page, so that the proper measurements may be taken off for actual use. A fact worth considering. All these examples are in English bond, but are good for any other bond.

Having dealt with foundations and footings, as we hope, in a satisfactory manner, it will not be out of place to say a few words on damp courses and means of preventing damp from getting up into the walls of buildings.

DAMP COURSES

In the construction of walls for dwellings, or in fact any other building of importance, it is essential that damp be prevented from being drawn up into the body of the wall by attraction; and the first thing to do in this case is to give some careful consideration to the floors, walls and footings of the cellar. Much has been written on the subject, and many recommendations of more or less value made as to the means of its prevention. Whether or not many of these are expediencies and not cures, the conditions in each case must decide.

All building materials, with perhaps the exception of granite, are porous and capable of absorbing and transmitting moisture in large quantity. The dampness in a building, however, arises from a variety of causes;

from absorption of moisture from the soil in or on which the building stands (a clay soil being peculiarly bad in this respect); from imperfect joints at window sills and lintels, as also unfilled and unpointed joints on the face of the wall; from moisture, forced into the walls during heavy driving rain storms; and from the water used in the process of construction, in the mortar and plaster, the wetting of brick, etc.

Every damp-preventing device, therefore, should be twofold in nature; it should, first, preclude the moisture from getting into the walls, and second, should not hinder it from getting out of the walls. The former is to be accomplished by an absolutely waterproof covering, such as asphalt or tar, or the complete isolation of the wall from any sources of dampness (exception, of course, being made here to the moisture which is *put* into the walls in building, and which should be allowed a proper opportunity to dry out). The latter is assured by the perfect ventilation of the walls on all sides.

The remedies for the dampness arising from the several causes above noted will be studied in their proper relative places.

There are many devices for keeping moisture from entering walls, and they may be divided into applications to the outside of the wall, and constructive devices.

Concerning cement as a guard against water, opinions now differ. That it is an excellent protective covering when it is well and thoroughly applied is not to be questioned. It is, however, frequently fractured by the settlement of the walls, and, being to some degree porous, suffers from the action of the frost. In either case it has no further value as a protective.

Of constructive devices to guard against dampness we have, first, those that are in the wall itself, and comprise the horizontal damp courses, hollow brick lining and facing and hollow wall.

Dampness will sometimes be found to arise from the soil below the floor, and in building upon such soils the whole site should be covered in, beneath the lowest floor, with dry earth, or, better still, with a thin layer of concrete, which will prevent the damp rising from that source.

In outside walls dampness will sometimes show itself in small patches here and there, and sometimes in quite large patches. The small patches probably arise from a few bricks of inferior quality which have inadvertently been built in the wall, and a cure can generally be brought about by covering the space on the inside of the wall beneath the paper with lead foil, using it to cover a space about 6 in. beyond the actual space of dampness.

The horizontal damp courses are of several kinds, and are placed at the bottom of the wall either on top of the footings or a short distance above them. The most effective course is one of asphalt or tar, applied in coats in the same manner as described for the facing of the walls. A greater degree of efficiency is given by laying the course of bricks immediately above the damp course, while the last coat is still hot and soft. When this damp course is set in a stone wall it would be better to lay a course of bricks and on this place the asphalt course, starting the stonework above the latter. A layer of slate, set in cement, has been much employed as a damp course. It has, however, the disadvantage of being very liable to fracture under uneven pressure.

Sheet lead is a most excellent damp course, and has been applied to the purpose for two centuries. For ordinary work its cost precludes its use.

It is claimed that the penetration of moisture can be hindered by building the wall so that there are no continuous bed joints through the wall. This device is presented on its own merits, the writer having no personal knowledge of its efficiency.

Another excellent damp course is found in the use of perforated terra cotta bricks. These are made the same size as the ordinary brick, and can, therefore, be readily bended into the wall. A course may be set immediately above the footings and another at or near the top of the wall. The bricks should be laid so that the openings run through the wall and so allow of ventilation and evaporation of any moisture that might rise in the hollow bricks themselves. The perforated bricks are also used to form a vertical damp course. They may be placed either on the inside or outside of the wall and may be laid as stretchers, as there is not the same liability to collect and retain moisture as there is in the horizontal course. Headers should be placed at frequent intervals to bond the facing to the body of the wall.

A simple and somewhat inexpensive system of rendering walls absolutely damp-proof and of adding very much to their strength and stability is to build the brickwork in two 4½-in. thicknesses with a ½ or ¾-in. cavity kept clear of mortar. Thin boarding is inserted in the cavity as the work advances, the space being afterwards filled with rock asphalt compositions. The compositions answer the double purpose of binding the two thicknesses together and making the wall impervious to moisture.

As a rule damp-proof courses should be 6 in. or more above the level of the external ground, but, where possible, under the wall plate carrying the joints for the floor.

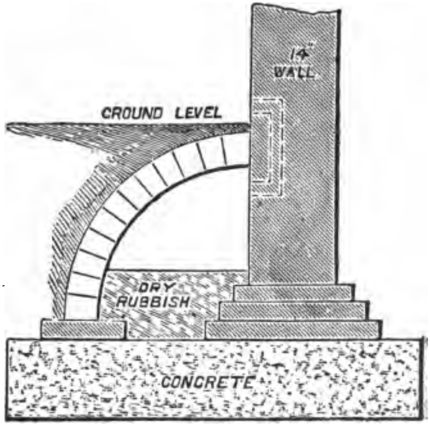


FIG. 29.

In buildings finished with a parapet wall, a damp-proof course should be inserted just above the flashing of the gutter, so as to prevent the wet which falls upon the top of the parapet from soaking down into the woodwork of the roof and into the walls below.

In some localities damp-proof courses are formed with slates set in cement; these are rather liable to crack, and thin impervious stones are better. Sheet lead has been used for the same purpose, and is most efficacious, but very expensive.

Arches are frequently rendered all over the extrados with asphalt or cement to prevent the penetration of wet, same as shown in Figs. 29-30. In addition to the precaution adopted to prevent damp out of the ground from rising in walls, it is necessary (especially when using inferior bricks or porous stone) to prevent moisture falling upon the outer face from penetrating to the interior of the wall.

The wet may be kept out of the interior of the wall by rendering the exterior surface with cement, covering it with slates fixed on battens or with glazed tiles set in cement; glazed or enameled facing brick answer the same purpose.

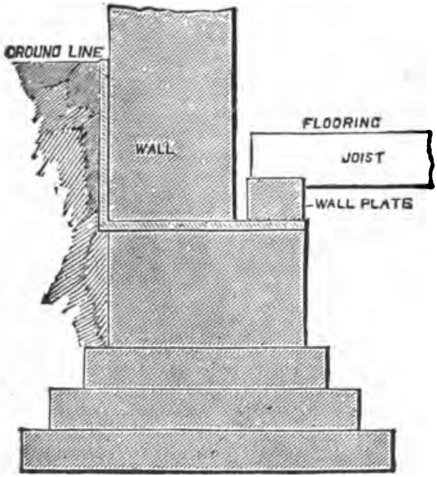


FIG. 30.

Sometimes vertical damp courses are used as shown in Fig. 30, particularly when the ground outside is higher than the wall plate inside, to prevent the damp penetrating through the wall. It will be seen that the damp course is bedded in the wall directly under the wall plate; this prevents the damp rising and destroying the wood. The vertical damp course acts in a similar manner in excluding the damp through the side of the wall; the joints of brickwork should be raked out to receive this damp course. Fig. 30 shows a good method of keeping damp out of the main walls. When the ground level is higher than floor level it will be seen that a 4½-in. wall is carried up to the ground level and covered on top with a stone coping fitted with an iron ventilating grating. By this method, as the damp penetrates through the 4½-in. outer wall, it rises and passes through the grating and into the open air. This wall is carried about 4½ in. from the face of main wall, and bonded into main wall as shown. Where the bonds enter, the main wall is tarred to prevent any damp entering.

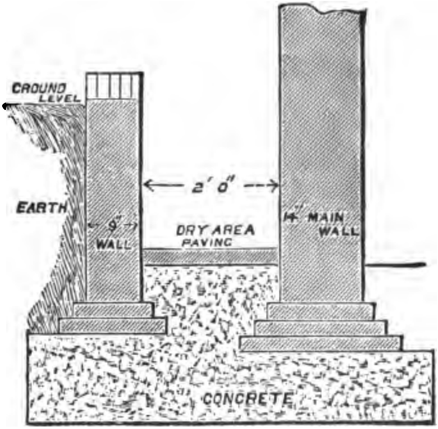


FIG. 31.

Another method of preventing damp from getting into a wall is to adopt what is known as the "dry area method," which is simply the building of a dwarf wall all around the building and leaving a space of two or more feet between the dwarf wall and the walls of the building as shown in Fig. 31. It will be seen by sketch that the ground is excavated to a width of 2 ft. from main walls and the dwarf wall built as shown to keep the water away. This area is necessary in damp situations, as any moisture or wet is carried away by a drain that is laid under the area, thus keeping the main structure dry. The dwarf wall is finished with a brick-on-edge coping built in cement. The floor of area is usually covered with cement concrete paving to prevent the water soaking in. Fig. 29 shows an enclosed dry area formed by means of the arch; this area is drained and the moisture is carried through the flue into the open air. This flue is lined either by neat cement or by asphalt to prevent the moisture penetrating into the wall. Hollow or cavity walls should be used for external work in damp situations exposed to driving rains. Such walls are of brick or stone, with a cavity of 2 or 2½ in. The external wall should be 4½ in., the thicker portion being inside; false headers being used in the outer wall. The thick wall inside will carry the doors and roofs, the woodwork being kept clear of the outer portion, which is liable to be damp.

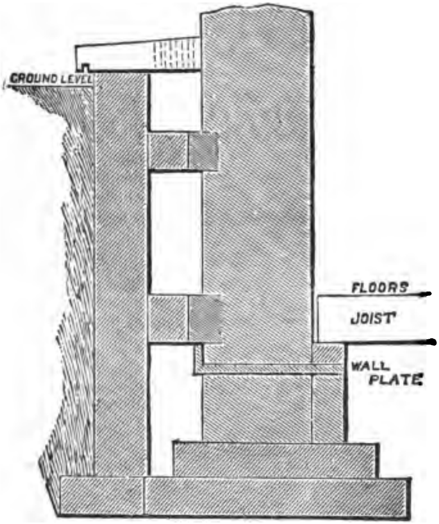


FIG. 31a.

The cavities should be ventilated by air-bricks in the external portion at top and bottom. Care must be taken that no mortar or other drippings get into them; movable boards or hay bands should be used.

The wall ties, generally of cast or wrought iron, galvanized or well tarred and sanded, should be employed to tie the two walls together. Walls constructed after this method not only exclude the damp, but the layer of air they contain, being a non-conductor of heat, tends to keep the building warm. Such walls are formed in two separate portions, standing vertically parallel to one another, and divided by a space of about 2 to 3 in.

Properties of Concrete.—A knowledge of the properties of materials is the first requisite for safe and economic designing of structures. The properties of reinforced concrete comprise not only those of the concrete and of the steel elements considered separately, but may be said to include those properties or characteristics of the composite mass that control the distribution of stresses between the elements of the combination of units and determine the nature of their inter-relation.

Compressive Strength.—Concrete is more often used in compression than in any other way, since it is more economical and has heretofore been considered more reliable under compressive strains than under transverse or tensile strains. Until very recent years engineers and architects hardly gave serious consideration to the value of concrete as a material to resist bending or tensile stresses, but at the present time comparatively few hesitate to use it in beams and similar situations where it is partly subjected to tensile stress, and considerable number of eminent members of both professions have constructed works where the tensile strength of the concrete is taken advantage of. The best practice, where any tensile strains can occur, is to reinforce the section with steel. The two chief factors that determine the compressive strength of a concrete are its age and the proportion of sand to cement in the matrix. The quality of the cement, sand, and aggregate have more or less influence on the resulting concrete, but with any good brand of modern high-burned Portland cement, clean sand, and clean, hard stone, substantially the same results may be secured. Factors of far greater weight are the manner and conditions of mixing and placing, and the personal equation of the operator. On this account it is extremely difficult to harmonize or draw conclusions from the large number of isolated tests that have been made by independent investigators under widely varying conditions and often with different objects in view.

When intelligently reinforced with steel, concrete becomes a material suitable and economical for beams, floors, and long columns, tanks, reservoirs, conduits, and sewers; admirably adapted to arch construction, and often economical for dams and retaining walls. Even in concrete that is not subjected to tension or flexure it is often desirable to introduce steel reinforcement to prevent the occurrence of cracks due to shock or settlement, or other causes.

Physical Tests of Cements.—The mechanical operations attending the manufacture of cement, such as the mixing, burning and grinding of the raw materials, bear intimate relation to the final physical properties of the cement, and should be analyzed just as closely as the chemical compositions; but in this treatise it is out of place to discuss manufacturing operations. The manufacture of a cement is therefore to be assumed correct if a sample of it passes those physical tests which are made for the purpose of determining its acceptance or rejectance for use. These tests are of such a character that the results of all experimenters are comparable; but it is not necessary, although it is desirable, that these tests should furnish values of the strength of the material, values which might be used in designing engineering work.

Concrete has a compressive strength varying from say 600 pounds to the square inch—such as we see in a great many dry mixed building blocks—to a strength as great as 4,000 pounds to the square inch. The latter strength was obtained with concrete bars made for testing purposes and cured under water. Concrete should have a compressive strength of fully 2,000 pounds to the square inch. The strength of concrete depends entirely upon how well the voids are filled with the cementing medium. Therefore, a lean mixture, thoroughly mixed, will oftentimes attain as great strength as a fatter mixture. A "fat" mixture varies from one part of cement to one part of sand, to a mixture of one part of cement, two parts of sand and four parts of small broken stone or clean gravel. Any concrete having a smaller proportion of cement is called a "lean" mixture.

The tensile strength of concrete is very much less than the compressive strength. Experiments made with mixtures of average proportions, give the ultimate fiber stress as one-eighth the breaking strength in comparison.

Sand.—Sand is the most widely distributed substance in nature, not only in the mineral but also in the animal and vegetable kingdoms. Clay contains no silica (the chemical name for sand). Sand is the siliceous particles of rocks containing quartz, production by the action of rain, wind, wave and frost. Some kinds of sand are also found inland; the deposits mark the sites of ancient beaches or river beds. Sand is classed under various heads, viz., calcareous, argillaceous and metallic. Sand varies in color according to the metallic oxides contained in them. Few substances are of more importance than sand for plastic purposes. Its quality is of primary importance for the production of good coarse stuff, setting stuff, and for gauging with Portland or other cements used for plaster work. Its function is to induce the mortar or cement to shrink uniformly during the process of setting, hardening or drying, irregular shrinkage being the general cause of cracking. Sand is also a factor in solidity and hardness; while being of itself cheaper and used in a larger proportion than lime or cement, it decreases the general cost of materials. There are three kinds—pit, river and sea sands. They generally contain more or less impurities, such as loam, clay, earth and salts, necessitating their being well washed in water, more especially for the finishing coats of plaster or cement work. Pit sand is sometimes found quite clean; it is generally sharp and angular. River sand is fine grained, not so sharp as pit sand, but makes good setting stuff. Sea sand varies in sharpness and size, and for plastering it should be washed to free it from saline particles which cause efflorescence.

As to the quality of sands, they are of very wide variety—so much so, that 1 part of an inferior or soft clayey sand will reduce the strength of mortar as much as 3 or 4 parts of clean sharp granitic sand. This is well exemplified in the sand test, which is made with what is called standard sand, being a pure silecious sand sifted through a sieve of 400 holes to the square inch and retained on one of 900.

Fine-grained sand is best for hydraulic lime; the coarse-grained is best for Portland cements and for floating.

Determination of Voids.—To determine the voids in the sand, or the material to be used as an aggregate, what is known as the "water test" is employed. In preparing for this test the sand or gravel must be perfectly dry. Sand has a greater volume when wet.

A receptacle holding a known amount, such as a quart jar, is filled with the material to be tested, sand, for example, and into this receptacle is poured as much water as the sand or other material will absorb. The water should be measured. The amount of water absorbed indicates the voids, and also indicates the exact amount of cement which it is necessary to use in order to produce a solid concrete.

Average sand will absorb 25 to 35 per cent of water, indicating from 25 to 35 per cent of voids, also indicating that the proportions of one part of cement to from three to five parts of sand are required to make a solid concrete.

The proper selection of sand and aggregate material is important. Care should be taken that the particles vary so in size as to reduce the voids to the smallest amount possible. With this careful selection the amount of cement required to produce good work is greatly reduced.

The voids in broken stone, sand and gravel or other materials used in making concrete vary from 33 per cent to 50 per cent of the entire mass. In order to get a good mixture it is generally figured that the stone contains 50 per cent of voids and that the sand contains 50 per cent of voids—the cement to fill the voids in the sand and be slightly in excess.

On this basis the theoretically perfect mixture, in which stone is used, is 1-2-4, for the reason that the sand and stone are generally mixed loose and the voids will, therefore, be about 50 per cent. If shaken in the measuring box, or slightly tamped, the proportion of voids can be reduced to about one-third of the mass.

A mixture of 1-3-5 may be water tight also if properly mixed and thoroughly tamped, but that again depends upon the porosity of the stone used.

Hydraulic Limes.—Hydraulic limes are those which have the property of setting under water or in damp places, where they increase in hardness and insolubility. The blue lias lime formation is that from which hydraulic lime is principally made. This lime, while it has excellent hydraulic properties, can hardly be classed as a cement. The stones which produce these limes contain carbonate of lime, clay, and carbonate of magnesia. The clay plays an important part in giving hydraulicity to the lime, consequently this power is greater in proportion to the amount of clay contained in the lime. The proportion of clay varies from 10 to 30 per cent. When lime contains clay it is not so easily slaked as pure lime, and does not expand so much in doing so, and therefore does not shrink so much in setting.

Lias lime (called blue lias from the color of the stone from which it is produced) is very variable in quality and is generally of a feeble nature, but is sometimes of an hydraulic nature. M. Vicat divides them into three classes: feebly hydraulic, ordinary hydraulic, and eminently hydraulic. "Those belonging to the first class contain from 5 to 12 per cent. of clay. The slaking action is accompanied by cracking and heat. They also expand considerably, and greatly resemble the fat limes during this process. They are generally of a buff color. Those of the second class contain from 15 to 20 per cent. of clay. They slake very sluggishly in an hour or so without much cracking or heat, and expand very little. They set firmly in a week. The eminently hydraulic limes contain from 20 to 30 per cent. of clay, are very difficult to slake, and only do so after a long time. Very frequently they do not slake at all, being reduced to a powder by grinding. They set firmly in a few hours, and are very hard in a month."

Retempered Mortar.—It is common to see a clause in specifications forbidding the use of mortar that has commenced to set. Today, however, it is well known that retempered mortar has a value. If allowed to set for about half an hour and then worked very vigorously for ten or fifteen minutes the time of ultimate set is prolonged.

There is no decrease in strength until after the second or third re-working. There is sometimes a slight loss in adhesiveness. This is restored by adding 10 or 15 per cent. of hydrate of lime to the mortar.

It is a difficult matter to make fresh concrete join to old work. The ordinary way is to clean the old surface and thoroughly wet it. Then put the new concrete against it.

The old way of mixing concrete was to put in just as little water as will hold the cement-mortar together. This is what is called the "dry" method of mixing concrete. In this method, if a handful of concrete is taken up and squeezed it will retain its shape when the hand is opened; but will not be wet in appearance and will not stain the hand. Such concrete is generally deposited in place and tamped very thoroughly until the excess of moisture is driven to the top and the mass becomes quakey.

It is surprising to see how wet concrete will become under thorough tamping when to the eye it looks perfectly dry before being put in place.

In a foundation or other structure where the strain is chiefly compressive, the surface of the concrete laid on the previous day should be cleaned and wet, but no other precaution is necessary. Joints in walls or in other locations liable to tensile stress are coated with mortar, which should be richer in cement than the mortar in the concrete, proportions 1:2 being commonly used.

Some engineers spread the cement dry upon the wetted surface of the old concrete, while others make it into a mortar; the latter method is necessary in many cases to seal the joints between the top of the old concrete and the bottom of the raised forms.

Removal of Forms.—Concerning the length of time that forms should be allowed to remain in place, the following instructions are given by the Aberthau Construction Company:

"Walls in mass work: one to three days, or until the concrete will bear pressure of the thumb without indention.

Thin walls: in summer, two days; in cold weather, five days.

Slabs (floors) up to 6 feet span: in summer, six days; in cold weather, two weeks.

Beams and girders and long spans: in summer, ten days or two weeks; in cold weather, three weeks to one month. If shores are left without disturbing them, the time of removal of the sheeting in summer may be reduced to one week.

Column forms: in summer, two days; in cold weather, four days, provided girders are shored to prevent appreciable weight from reaching columns.

Conduits: two to five days, provided there is not a heavy fill upon them.

Arches: of small size, one week; for large arches with heavy load, one month."

"These intervals of time are only approximate. The proper length of time for the removal of forms depends, in the main, upon the temperature, and amount of moisture in the atmosphere. The suggestion has been made that two or three times a day a sample of concrete be taken from the mixer and allowed to "set" on the ground under the same conditions as the construction until the time is up for the removal of the forms. These sample specimens may then be placed in a testing machine to determine whether or not the actual strength of the concrete is sufficient to carry the dead, and construction loads.

Even this plan does not provide for the possibility of an occasional poor batch of cement, so that watchfulness and good judgment must also be exercised. Beams and arches of long span require to be supported for a longer period than do short spans, for the reason that the dead load is proportionally large, and therefore the compression in the concrete is large even before the live load comes upon it.

Even in summer, during a damp, cloudy period, wall forms cannot sometimes be removed inside of five days, with other members in proportion. Occasionally batches of concrete will set abnormally slow, either because of slow-setting cement, or impurities in the sand, and the foreman, and the inspector must watch very carefully to see that the forms are not removed too soon. Trial with a pick may assist in reaching a decision."

Reinforced Concrete.—With the advent of modern concrete the facilities with which reinforcing rods or bars of metal may be embedded anywhere in the mass of the masonry was soon seen and taken advantage of. The compressive resistance of concrete is about ten times its tensile resistance, while steel has about the same strength in tension as in compression. Volume for volume steel costs about fifty times as much as concrete. For the same sectional areas steel will support in compression thirty times more load than concrete, and in tension three hundred times the load that concrete will carry. Therefore, for duty under compression only, concrete will carry a given load at six-tenths of the cost required to support it with steel. On the other hand, to support a given load by concrete in tension would cost about six times as much as to support it with steel. If the various members of a structure are so designed that all of the compressive stresses are resisted by concrete and steel is introduced to resist the tensile stresses, each material will be serving the purpose for which it is the cheapest and best adapted and one of the principles of economic design will be fulfilled.

Other important advantages secured in the combination of concrete and embedded steel are that the protection of the metal elements from corrosion is practically perfect; that, with properly selected ingredients, the fire and heat resisting qualities are very high, perhaps surpassed by no other building material except fire-brick.

TABLE 2.—RANSOME BARS.

Size of Bar	Weight to the Foot	Elastic Limit	Ultimate Strength
1¼ inch square	5.312	55,450	83,150
1 inch square	3.400	55,760	84,730
⅞ inch square	2.603	56,150	84,730
¾ inch square	1.913	56,720	85,240
⅝ inch square	1.328	57,890	85,820
½ inch square	.850	60,120	86,350
⅜ inch square	.478	61,800	86,600
¼ inch square	.213	62,350	86,700

The Strength of Concrete.—The strength of concrete varies (1) with the quality of the materials; (2) with the quantity of cement contained in a cubic yard of the concrete; and (3) with the density of the mixture.

We may say that the strongest and most economical mixture, consists of an aggregate comprising a large variety of sizes of particles, so graded that they fit into each other with the smallest possible volume of spaces or voids, and enough cement to slightly more than fill all of these spaces or voids between the solids of the aggregate. It is obvious that with the same aggregate the strongest cement will make the strongest concrete.

On important construction the various materials to be used should be carefully tested, and specimens of the mixture selected made up in advance and subjected to test. As a guide to the loads which concrete will stand in compression—that is, under vertical loading where the height of the column or mass is not over, say, 12 times the least horizontal dimension—we may give the following approximate figures as safe strengths, after the concrete has set at least one month, for the proportions which have previously been selected in this section as typical mixtures.

The figures, compared with the results of recent experiments on 12-inch cubes, allow a factor of safety of six at the age of one month, or eight at the age of six months, and are based on conservative practice.

SAFE STRENGTH OF PORTLAND CEMENT CONCRETE IN DIRECT COMPRESSION.

Proportions	Pounds per square in.	Tons per square ft.
1:2:4	410	29
1:2½:5	360	25
1:3:6	325	23
1:4:8	260	18

With a large mass foundation, take values one-eighth greater.
With a vibrating or pounding load, take one-half these values.

TRIANGULAR REINFORCEMENT 4-INCH MESH, WITH 12 ½ GAUGE CROSS WIRES.

SIZES OF WIRE, AREAS AND TENSIL STRENGTH PER SQUARE FOOT, AND PER FOOT WIDTH.

WEIGHTS PER 100 SQUARE FEET. STR.—MINIMUM ULT. STRENGTH.

Size of Wire Longitudinals		One Wire each Longitudinal						2 Wires each Longitudinal						3 Wires each Longitudinal					
		Longitudinal Members	Cross Wires 12½ Gauge	Total per square foot	Cross section per ft. width	Style Numbers	Weight per 100 sq. ft.	Longitudinal Members	Cross Wires 12½ Gauge	Total per square foot	Cross section per ft. width	Style Numbers	Weight per 100 sq. ft.	Longitudinal Members	Cross Wires 12½ Gauge	Total per square foot	Cross section per ft. width	Style Numbers	Weight per 100 sq. ft.
¼ In	Area	.1472	.0381	.1853	.1701														
	Str.	9420	2440	11860	10886	23	65.8												
No. 4.............	Area	.1192	.0381	.1573	.1421			.2384	.0381	.2765	.2612			.3576	.0381	.3957	.3804		
	Str.	7628	2440	10068	9094	24	56.7	15257	2440	17697	16716	31	96.7	22886	2440	25324	24345	38	135.8
No. 5.............	Area	.1009	.0381	.1390	.1237			.2018	.0381	2399	.2246			.3027	.0381	.3408	.3255		
	Str.	6457	2440	8897	7916	25	50.7	12915	2440	15353	14374	32	84.5	19372	2440	21812	20832	39	117.5
No. 6.............	Area	.0868	.0381	.1249	.1096			.1737	.0381	.2118	.1965			.2605	.0381	.2986	.2833		
	Str.	5555	2440	7995	7014	26	45.8	11116	2440	13556	12576	33	75.0	16672	2440	19112	18131	40	103.4
No. 8.............	Area	.0618	.0381	.0999	.0846			.1236	.0381	.1617	.1464			.1854	.0381	.2235	.2082		
	Str.	3955	2440	6395	5414	27	37.6	7910	2440	10350	9369	34	58.5	11865	2440	14305	13324	41	78.7
No. 10.............	Area	.0429	.0381	.0810	.0657			.0858	.0381	.1239	.1086			.1287	.0381	.1668	.1515		
	Str.	2745	2440	5185	4204	28	31.3	5491	2440	7931	6950	35	45.9	8236	2440	10676	9696	42	60.1
No. 12......	Area	.0259	.0381	.0640	.0487			.0519	0381	.0900	.0747			.0778	.0381	.1159	.1006		
	Str.	1657	2440	4097	3116	29	25.6	3321	2440	5761	4780	36	34.7	4979	2440	7419	6438	43	43.4
No. 12½...........	Area	.0228	.0381	.0609	.0456			.0457	.0381	.0838	.0685			.0685	.0381	.1066	.0913		
	Str.	1459	2440	3899	2918	30	24.4	2924	2440	5364	4384	37	32.4	4384	2440	6824	5843	44	40.0

COMPARATIVE STRENGTH OF GREY LIME AND PORTLAND CEMENT MORTAR, ALSO PORTLAND CEMENT MORTAR WITH THE ADDITION OF LIME AND MORTAR.—REDGRAVE.

No.	No. of Tests	Proportions				Breaking Strain on 2.25 square inch in lbs.	Breaking Weight per square inch in lbs.	Ratio as compared with Lime Mortar	Ratio as compared with Cement Mortar	Remarks
		Sand	Cement	Lime	Water					
1	17	2.00	....	1 00	1.33	61.06	27.13	....	....	Three samples
2	27	2.00	....	1 00	1.33	106.07	47.09	....	....	Grey lime.
3	27	2.00	....	1 00	1.33	82.00	36.44	....	....	Water includes that required for slaking lime.
1	15	6.00	1.00	....	1.25	233.53	103.79	2.81 to 1	....	Cement taken from bulk in store.
2	20	8.00	1.00	....	1.66	154.80	68.80	1.86 to 1	....	
3	35	10.00	1.00	....	2.00	112.88	50.16	1.36 to 1	....	
1	70	6.00	1.00	0.50	1.50	165.31	73.47	2.00 to 1	0.70 to 1	Water includes that required for slaking lime.
2	74	8.00	1.00	0.66	2.00	132.62	58.94	1.60 to 1	0.85 to 1	
3	85	10.00	1.00	0.83	2.50	95.27	42.34	1.14 to 1	0.84 to 1	
				Loam.						
1	21	6.00	1.00	0.50	1.00	136.80	60.80	1.64 to 1	0.58 to 1	Yellow loam, fresh dug, and rather damp.
2	25	8.00	1.00	0.66	1.33	86.48	38.43	1.04 to 1	0.55 to 1	
3	19	10.00	1.00	0.83	2.00	64.50	28.66	0.77 to 1	0.57 to 1	

TABLE 3.—SAFE LOAD FOR MASONRY FOOTING COURSES—KIDDER.

Kind of Footing.	R* in pounds to the square inch	Offset for a pressure to the square foot on the bottom of the course, multiply thickness of course by the proper factor.					
		0.5 Ton	1 Ton	2 Tons	3 Tons	5 Tons	10 Tons
Bluestone flagging......	2700	3.6	2.6	1.8	1.5	1.2	0.8
Granite	1800	2.9	2.1	1.5	1.2	1.0	0.7
Limestone	1500	2.7	1.9	1.3	1.1	0.9	0.6
Sandstone	1200	2.6	1.8	1.3	1.0	0.8	0.5
Slate	5400	5.0	3.6	2.5	2.2	1.5	1.2
Best hard brick.........	1200	2.6	1.8	1.3	1.0	0.8	0.5
Concrete { 1. Portland cement 2. Sand ..... 3. Pebbles	150	0.8	0.6	0.4	...	...	...
Concrete { 1. Rosendale cement 2. Sand .... 3. Pebbles	80	0.6	0.4	0.3	...	...	...

* Modulus of rupture, values given by Prof. Baker in "Treatise on Masonry Construction."

ERECTION OF BUILDINGS.

Heavy Timber Framing.—Since the introduction of steel frames into building construction, the use of timber frames in roofs, buildings, bridges and trestle work has greatly fallen off, particularly in or near large cities, where timber has become scarce and costly, but in the west, north, and south, timber structures are often made use of, and will be for many decades yet. Indeed, even when steel is made use of timber has to be frequently employed in special cases; so that a knowledge of framing is as necessary to the general workman to-day as it ever was. Especially does this apply to the man who desires to become a competent millwright.

Heavy framing is an art that requires considerable ability and intelligence on the part of the operator, inasmuch as it is not one of those branches of the trade where the "cut and fit" process can be applied.

Each piece of timber, whether it be a girt, a chord or beam, a post, brace, sill, girder, strut or stringer, must be dealt with, and given its proper shape, length and relationship to the part or parts it is to be connected with, without its being brought in direct contact with it until all is ready to be put together and pinned up solid. A clear head and a good memory, along with the faculty of exactness, are absolutely necessary qualifications for the making of a good framer. He must see to it that all tenons are the right size to suit the mortises which they are intended to fill, and that all mortises are clearly and smoothly finished and not too large or too small to snugly receive the tenons, and all this must be done without any "trying and fitting." The charm of good framing lies in the fact that every mortise and tenon must be "driven home" with a heavy wooden mall; but tenons should not fit so tight as to require more than ordinary driving.

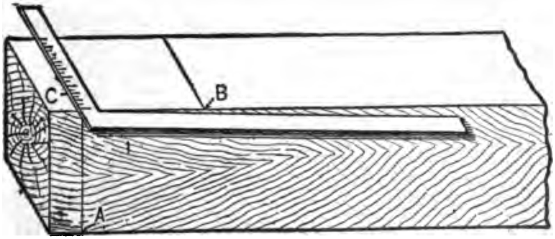


FIG. 32.

Following an authority on the subject of laying out work by the steel square "the ends of the mortise are first struck as indicated at A and B, Fig. 32, and while the square is in the position indicated the mark C is made for the working side of the mortise, which is always the narrower side unless the two are equal. In practice it is best to mark the cut-off at the end of the timber first, or if it does not need cutting off, place the square over the end of the stick, and mark back along the blade the 1½, 2 or 3 inches required for the shoulder. This makes sure that there is no projecting corners to give trouble later on.

If a tenon is being struck the same method is followed, going entirely around the stick but working in both directions from the face corner. The ends of the mortise or shoulder of the tenon being thus treated, the sides are marked by reversing the square, placing the inside of the blade at E, Fig. 33, fair with the mark C previously made, and taking the same distance—in this case 2 inches—on the tongue of the square, as shown at B. Now by holding the square firmly with the thumb and fingers of the left hand both sides of the tenon can be marked, but great care is necessary to prevent the slipping of the square. If there is any wane on the stick it is hard to tell when the mark D is exactly in line with the vertical face of the timber, and this matter must be determined by sighting down the side of the stick. It is also necessary to drop the blade of the square a little further, as at B, when squaring across a "wany stick."

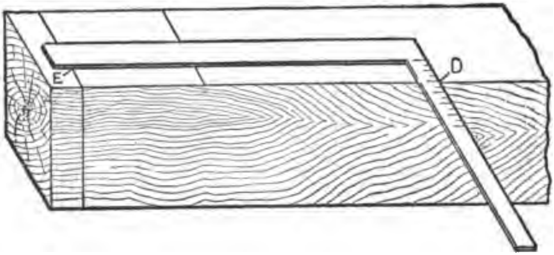


FIG. 33.

In every heavy timber framing a bridge framing steel square could be employed. These have a blade three inches wide and a tongue one and a half inches wide. The blade is used to lay out mortises and tenons of three-inch dimensions. There is a slot one inch wide cut down the center of the blade, the slot is twenty-one inches long and it may be used on one inside edge to make a two-inch mortise or tenon; this is done by using one outside edge and one inside edge. These squares are made by Sargent & Co., of New York, and cost from $2.50 to $5.00 each. The squares are very handy for bridge builders and for framing all kinds of heavy timber.

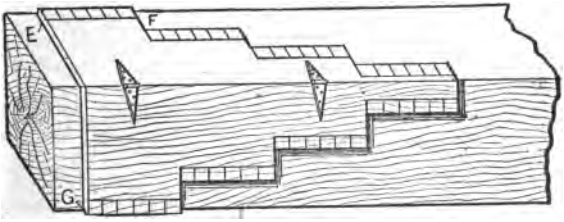


FIG. 34.

A kind of templet or guide is made use of sometimes, for laying out work, it is much handier, and easier to work with than the square, and will aid in laying out work much more rapidly. These templets are made in both wood and metal. They are hinged at the angle as shown in the sketches herewith, so they may work easily over wany edges or can be folded together and stowed away in a tool chest.

In order to lay out a mortise, slide the tool along until the end comes flush with the longest corner; then mark the end of the mortise, as at E of Fig. 34. At the same time mark the other end of the mortise, F, Fig. 34; then slide back the marker and strike that line after having first struck the line E. Next reverse the tool and select the width of shoulder required—2 inches in this case—and mark alongside the board on the timber. This fixes one side of the mortise or tenon, and a mark alongside the right width of tool, H, Fig. 35, finishes that mortise in very quick time." Apart from this description, the workman will find in making use of this tool many places where it can be employed to advantage. If the whole tool was constructed of metal, it would not cost any more than if made of wood, as described in the foregoing, and it would be neater, lighter, much more compact, and would last for all time.

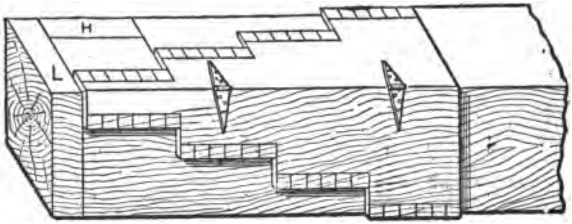


FIG. 35.

While it is true that this templet is a great help in rapid framing and, while in some cases where the timber is wany, or lacking on the arrises, something of the kind is necessary. Where the writer has met with wany timber he has often tacked a planed board on the side of the timber to be worked keeping the upper edge even with the top of the timber, then the square can be used for making over as the board forms a good surface to work the square from. When the templet is used, the necessity of the board is done away with, as the vertical portion of it takes the place of the board. The method of using the square for cutting rafters, braces, and other angular work, has been shown and described elsewhere, and need not be again referred to.

There is one matter in framing that should by all means be described and properly illustrated, and that is the question of "boxing." Some framers may not know what the term "boxing" means; but every "old hand" at the business has, no doubt, a vivid recollection of the term and its uses. "Boxing" in framing may be described as preparing a true real square with the jaws of the mortise for the shoulders of the tenon to butt solidly against. To accomplish this often requires the removal of portions of the timber before a flat square surface is found, and this may reduce the thickness of the timber operated upon. If we suppose four or five posts on the side of a building, and these posts are supposed to be 12x12 inches in section and in preparing these posts to receive the tenons it is necessary to remove over the face of each mortise one-quarter of an inch or more, and the girts or connecting timbers have their shoulders cut to suit the 12-inch posts, it will be seen that the length of the building at the line of girts will be less than intended. If forced into mortises made the proper distance apart in the sills, the outside posts will not be plumb and it will be found impossible to make the plates fit in place, as the mortises on the ends will be found too far apart, and this would lead to all sorts of trouble and vexation. In boxing, we suppose the posts to be, say 11½ inches instead of 12 inches. This allows half an inch for boxing, and this necessitates the girt between each set of posts, to be cut one inch longer between shoulders than if no boxing was prepared. In cases where posts are pierced on both faces and boxed, the post where the tenons enter, if directly opposite, may have to be reduced to 11 inches and must be accounted for on that basis. The young framer must be particular about the boxing and the necessary reduction of timbers when laying off his lengths of girts or

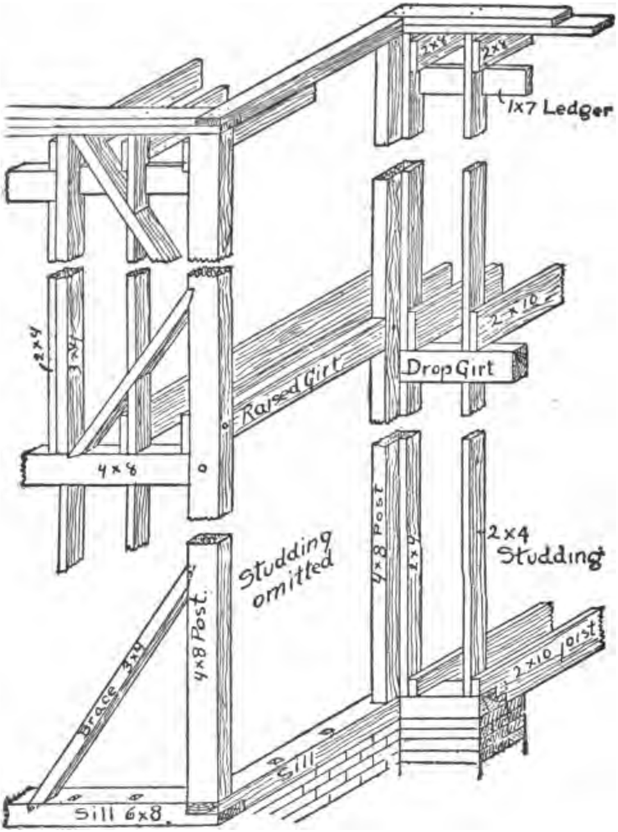


FIG. 36.

bracing timbers, if not he will be sure to get into trouble or botch the job.

Fig. 36 shows an example of heavy framing, which no doubt will enable the student to get an idea of how that style of work is done.

The illustration shown at Fig. 37 makes plain the method of constructing a double floor. The binder rests on a wall or posts. This makes a fine floor, and is in a measure sound proof.

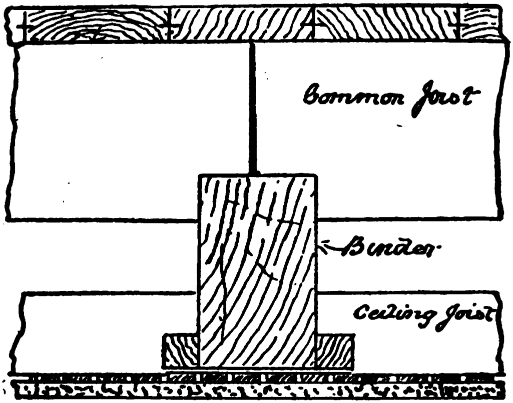


FIG. 37.

Figs. 38, 39, 40 and 41, which are borrowed from *Architecture and Building*—old series—will convey to the reader a number of excellent ideas as to the combination of iron and wood in floor framing.

Fig. 42 shows the manner usually adopted in preparing the floor timbers around a hearth, chimney breast, stair well hole, or openings for trap doors or similar work. The trimmers and headers are made with heavier timbers than the joists, and the tail beams are let into the headers with either plain or tusk-tenons. Tusk-tenons, of course, are the best, but entail much labor and care. A tusk-tenon with a run-over top is shown in Fig. 42a.

This makes a good clean joint for running over a girt or bearing timber, and can be nailed together over the joint as shown, thus holding the work so that it cannot spread. The tenon is shown at a.

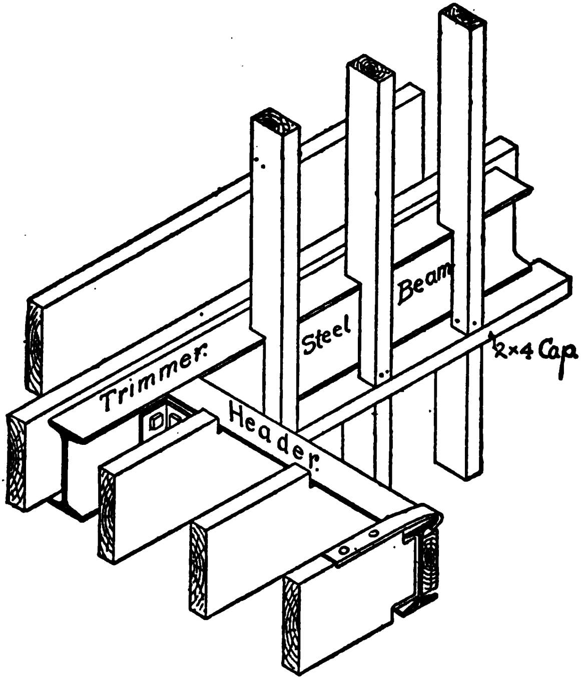


FIG. 38.

There are various shapes of tusk-tenons, some of which are shown in the following examples. Below is given a brief description of what is considered the best kind of a tenon:

The usual rule for cutting a common tenon is to make it one-third the width of the timber and this rule should be followed as far as possible in designing a tusk-tenon. The projection of the tenon from the beam out of which it is cut is called its root, and the surfaces immediately adjacent to its root on the sides are called the shoulders.

The tusk-tenon was devised in order to give the tenon a deep bearing at the root, without greatly increasing the size of the mortise. Making the mortise unduly large would, of course, weaken the girder. The desired deep bearing is secured by adding below the tenon a tusk having a shoulder which in trimmer work penetrates to a depth about one-sixth the thickness of the joist. Above the tenon is formed what is called a "horn," the lower end of which penetrates to the same extent as the tusk. By this arrangement the strength of the tenon is greatly increased as compared with the common form, while the mortise is not made very much larger. In order to hold the parts together the tenon is projected through the girder and pinned on the outside as shown in the sketches.

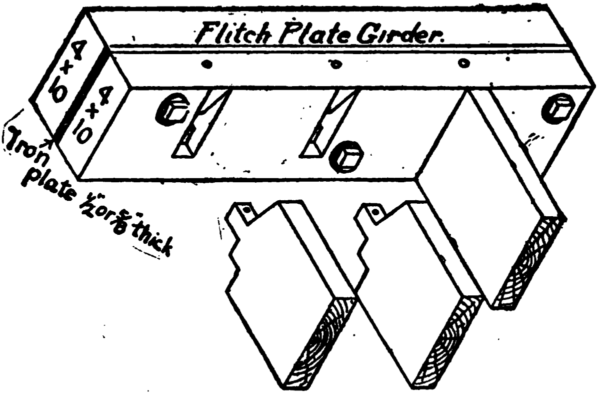


FIG. 39.

So much for a description of the tusk-tenon, as it is

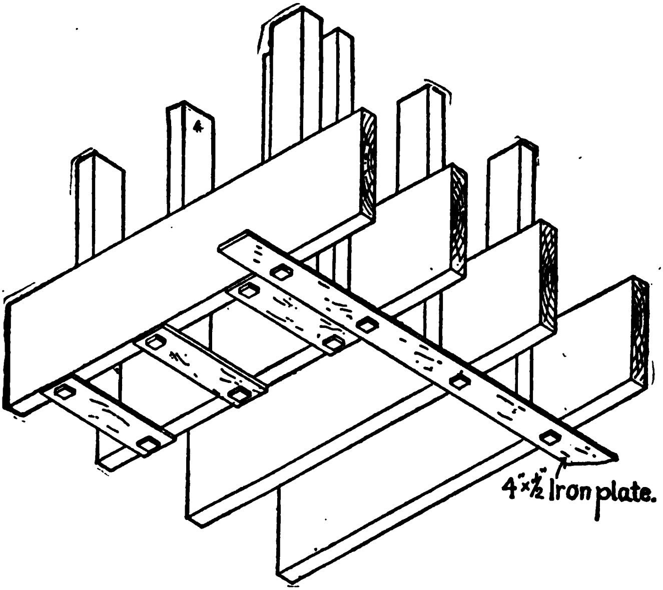


FIG. 40.

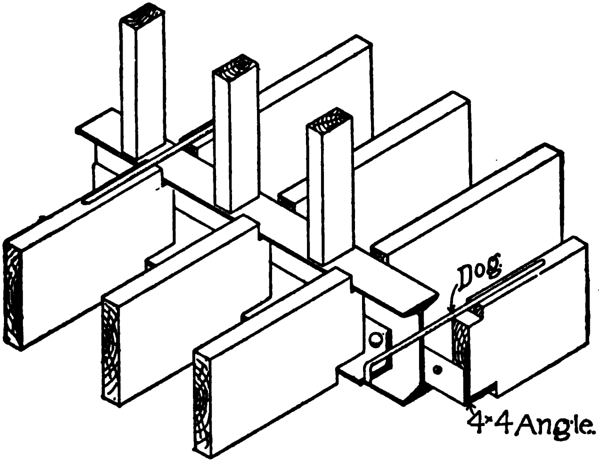


FIG. 41.

theoretically, and as illustrated in Fig. 42. Many times, however, the tusk-tenon is attempted upon the lines shown in Fig. 42a.

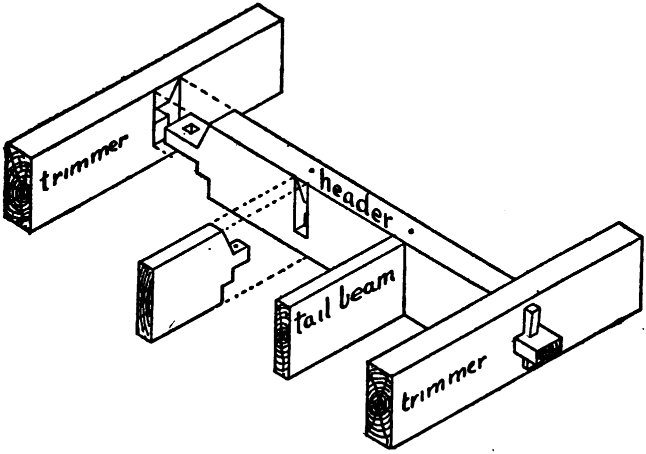


FIG. 42.

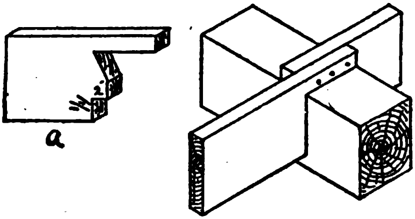


FIG. 42a.

Roof Framing.—A couple of good purlin roofs suitable for many places, are shown in Figs. 43 and 43a.

The one shown at Fig. 43 is known as a queen post truss, but having queen rods instead of posts. Two additional braces and one rod have been added to the members of the truss so as to take up the half load between the points F and H. According to the conditions of loading it has been formed sufficiently strong to bear all the load it may ordinarily be called upon to resist.

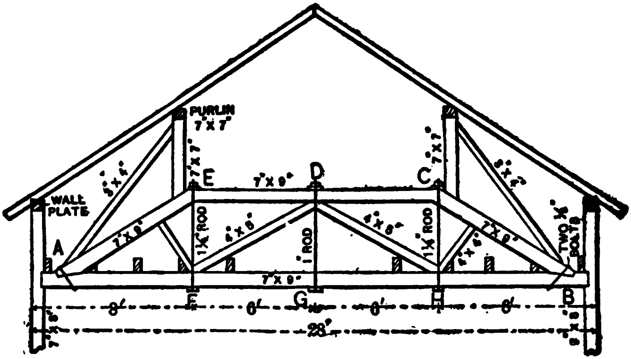


FIG. 43

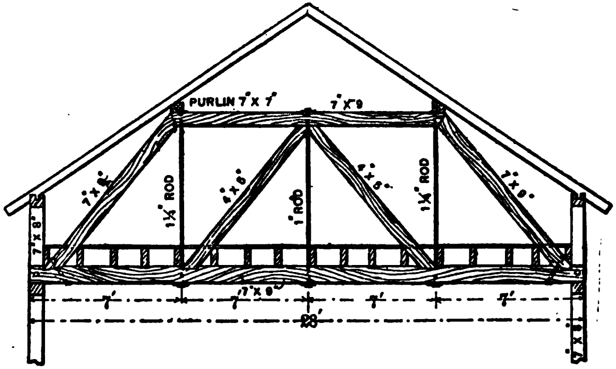


FIG. 43a.

Taking the roof load first and assuming 40 pounds per square foot, including wind, snow and weight of roof itself, it is found that a load of about 7,280 pounds will be concentrated at or near the points E and C. This load

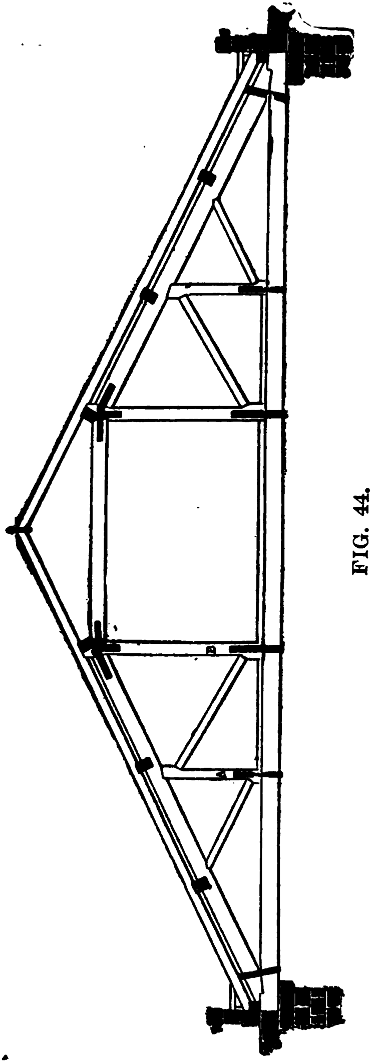


FIG. 44.

will cause a stress of about 13,500 pounds compression in each of the rafters A E and C B; also a compression strain of 11,300 pounds in the straining beam E C, as well as a tensile strain of about 11,300 pounds in the tie beam A B. In computing the strains due to the floor load, 200 pounds per square foot of floor area have been taken.

Fig. 44 shows a queen-post roof for a span of 60 feet. This truss is designed on the same principle as Fig. 43a, that is, with queen-posts B, and additionally strengthened by suspension post A. These are strapped up to the tie-beam by wrought-iron straps, made of 3/8x3-inch iron, bolted to the posts. The pitch of the principal rafter is less somewhat than over Fig. 43a.

The scantlings are as follows:

Principal rafters	11 x6	inches
Tie-beam	12½x6	inches
Queen-post B	8 x6	inches
Suspending-post A	3½x3½	inches
Struts (large)	4½x3½	inches
Struts (small)	3½x2½	inches

Fig. 45 shows a platform roof of 35 feet span. The tie-beam in this example is scarfed at a and b, and the center portion of the truss has counter-braces, c c. The longitudinal pieces, e e, are secured to the heads of the queen-posts, and the pieces d carry the platform rafters A. In this connection it may be of importance to the better understanding of the principles of strength entering into combination roof trusses to give Tredgold's rules for finding the proper dimensions of the timbers forming king and queen-post trusses, which are quite simple.

Rule.—Multiply the square of the length in feet by the span in feet, and divide the product by the cube of the thickness in inches; then multiply the quotient by 0.96 to obtain the depth in inches.

Mr. Tredgold gives also the following rule for the rafters, as more general and reliable:

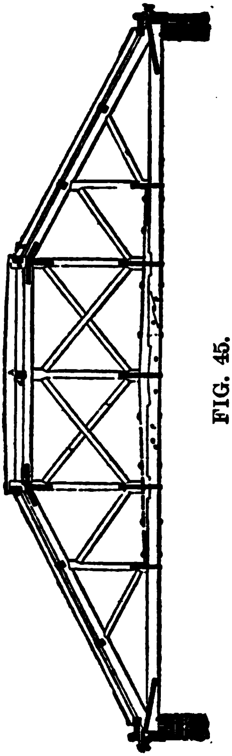


FIG. 45.

Multiply the square of the span in feet by the distance between the principals in feet, and divide the product by 60 times the rise in feet; the quotient will be the area of the section of the rafter in inches.

If the rise is one-fourth of the span, multiply the span by the distance between the principals, and divide by 15 for the area of section.

To find the dimensions of the queen-posts:

Rule.—Multiply the length in feet of that part of the tie-beam it supports; the product, multiplied by 0.27 will give the area of the post in inches; and the breadth and thickness can be found as in the king-post.

The dimensions of the struts are found as before.

To find the dimensions of a straining-beam:

Rule.—Multiply the square root of the span in feet by the length of the straining-beam in feet, and extract the square root of the product; multiply the result by 0.9, which will give the depth in inches. The beam, to have the greatest strength, should have its depth to its breadth in the ratio of 10 to 7; therefore, to find the breadth, multiply the depth by 0.7.

To find the dimensions of purlins:

Rule.—Multiply the cube of the length of the purlin in feet by the distance the purlins are apart in feet, and the fourth root of the product will give the depth in inches, and the depth multiplied by 0.6 will give the thickness.

To find the dimensions of common rafters, when they are placed 12 inches apart:

Rule.—Divide the length of bearing in feet by the cube root of the breadth in inches, and the quotient multiplied by 0.72 will give the depth in inches.

It may be well to note some practical memoranda of construction which cannot be too closely kept in mind in designing roofs.

Beams acting as struts should not be cut into or mortised on one side, so as to cause lateral yielding.

Purlins should never be framed into the principal rafters, but should be notched. When notched they will carry nearly twice as much as when framed.

The diagram, Fig. 46, shows the elevation of a king-post roof suitable for a span of 35 or 40 feet.

By the rules for calculating the sizes of timbers the dimensions will be found to be as follows:

A, Tie-beam	13 x5	inches
B, Principal rafters	8½x5	inches
C, Struts	4 x2½	inches
D, King-post	7½x5	inches

Fig. 47 is the design for a king-post roof, for a span of from 40 to 45 feet.

The purlins here are shown framed into the principals, a mode of construction to be avoided, unless rendered absolutely necessary by particular circumstances.

The scantling, as determined by the rules, is as follows:

Principal rafters	10 x5	inches
Tie-beam	11½x6	inches
King-post	7¾x6	inches
Struts	4 x2½	inches
Purlins	10 x6	inches

The principals being 10 feet apart.

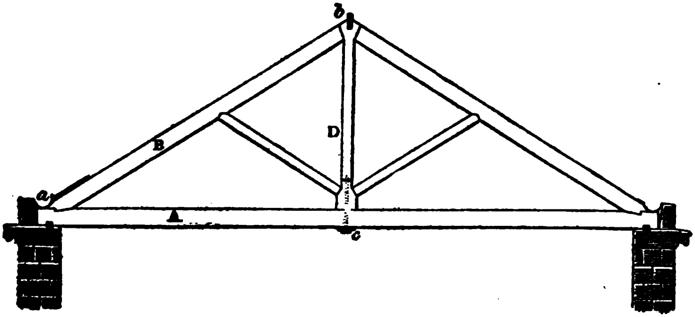


FIG. 46.

The following example (Fig. 48) is taken from Bell's Carpentry, and shows a strong roof, and one that will suit admirably for a factory or machine shop where there is likely to be jars or shakes caused by the machines in motion, or the rolling in of heavy freight. This roof may have a span of fifty feet, or even more if necessary. The principal rafter is set back a foot from the end of the tie-beam to give room for the wall-plate; the rise of the roof is 5 inches to the foot. In framing roofs of this kind the supporting rods should be furnished before commencing the frame; for then the length of the short principal rafters, and that of the straining beam can be regulated or proportioned according to the length of the rods. It is best, however, for the middle rod to be twice

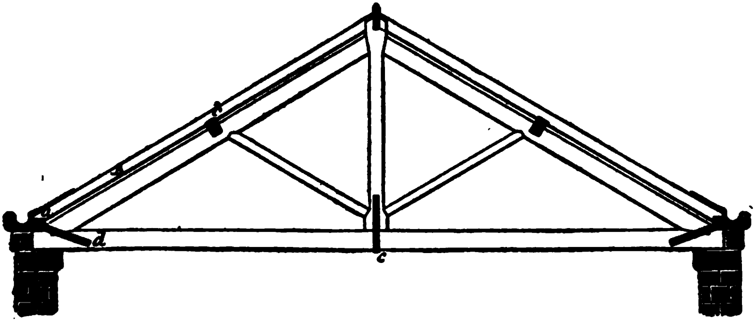


FIG. 47.

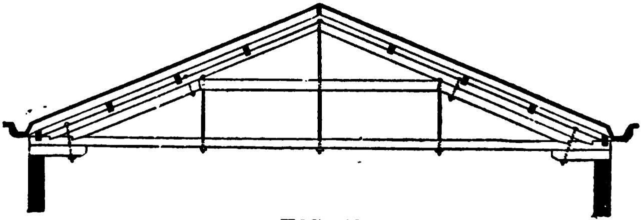


FIG. 48.

the length of the short ones, reckoning from the upper surface of the beam to the upper surface of the principal rafters, and allowing one foot more to each rod for the thickness of the beam, and the nut and washer. For example, the middle rod is 11 feet long and the short ones 6 feet each; which, after allowing 1 foot, as above mentioned, makes the length of the long one, above the work side of the beam, twice that of the short ones.

The length of the rod above the beam is the rise of the rafter, and the distance from the center of the rod to the foot of the rafter is the run of the rafter; the length of the rafter can, therefore, be found by the usual way.

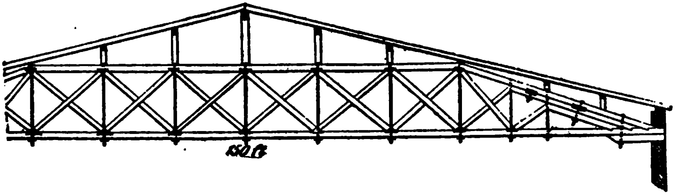


FIG. 49.

To find the length of the straining beam, add the run of the short principal rafter to the lower end bevel of the long one; substract this run from the run of the long principal, and the difference will be half the length of the straining beam.

The bolsters under the ends of the tie-beams are of the same thickness as that, and about 5 feet long.

The principle on which the truss shown in Fig. 49 is constructed is essentially the same as that of the Howe Bridge. The braces are square at the ends, the hardwood blocks between them being beveled and placed as shown in the diagrams. Each truss of this frame supports a purlin post and plate, as represented.

Following will be found a few general rules governing timber roof framing which the workman should always have in mind when building or designing a roof of any kind.

1. Every construction should be a little stronger than "strong enough."

2. Roofs should neither be too heavy nor too slight; both extremes should be rigorously avoided.

3. Flat-pitched roofs are not so strong as higher pitched ones.

4. Suitable pitches of roofs for various coverings are: Copper, lead, or zinc, 6 degrees; corrugated iron, 8 degrees; tiles and slates, 33 degrees to 45 degrees.

5. Approximate weight of roofs per square: The timber framing, 5½ cwt.; Countess slates, 6½ cwt.; add for 1-in. pine or hemlock boarding, 2½ cwt.; plain tiles, 14 cwt.; 7 lb. lead, 6 cwt.; 1-32 in. zinc, 1½ cwt.

6. The construction should be able to withstand an additional weight of 30 cwt. per square for wind pressure.

7. When the carpentry forming the roof of a building is of great extent, instead of being injurious to the stability of the walls or points of support, it should be so designed that it will strengthen and keep them together.

8. Forms of roofs for various spans should couple, up to 11 ft.; couple close, to 14 ft.; collar, to 17 ft.; king-post, to 30 ft.; queen-post, to 46 ft.; queen and princess, to 75 ft.

9. Roof trusses should be prepared from sound, dry timber, white or red pine, free from large knots, sap, and shakes, all parts to hold sizes shown in figured dimensions, and all joints to be stub-tenoned and to fit square to shoulders. Tie-beams should be cambered ½ in. in 10 ft., and straps and bolts be of best wrought-

iron. No spikes should be used in the construction except for fixing cleats.

10. Tie beams should be supported every 15 ft.

11. Struts should be taken as nearly as possible under bearing of purlin.

12. The straining beams in spans of 50 ft. and upwards require support, and a king bolt or post should be introduced.

13. To find the thickness of king-post trusses, divide the span by five and call the quotient inches. Assume 9 in. and 5 in. as the standard depth of tie-beams and principal rafter respectively for 20 ft. span; add 1 in. to each for every additional 5 ft. of span. King-posts and struts to be square.

14. To find the thickness of queen-post trusses, divide the span by eight and call the quotient inches; if odd parts result, add 1 in. for tiles, and for slates take off the fraction. Taking the standard depth of tie-beam and principal for 32 ft. span to be 11 in. and 6 in. respectively, add 1 in. to each for every 5 ft. of additional span. The struts and body of the queens to be made square.

15. Wall plates are used to distribute the weight of roof timbers, and also to act as ties to the walls. For this reason tie-beams should be cogged to the plates, the latter dove-tail-halved at the angle, and dove-tail-scarfed in longitudinal joints. Wall plates in roofs should be creosoted or otherwise protected against rot, and bedded in cement knocked up stiff.

16. Purlins should be cogged or notched on to principal rafters, and not framed between them. When cogged or notched they will carry nearly twice as much as when framed.

17. The available strength of tie-beams is that of the

uncut fibres, and, therefore, mortises should be shallow, and all notching be avoided.

18. Scarfs in tie-beams should be made between the points of support, and not directly under them, as any mortises or bolt-holes at these points reduce the strength of the beam.

19. Dragon ties should be provided at the angles of hipped roofs to take the thrust of the hips and to tie in the ends of wall plates. It is best that the hip should be deep enough to birdsmouth over the angle brace.

20. Wind braces, which are diagonal ties in roofs open at the ends, as in railway stations, to withstand the overturning or racking pressure of the wind, may be of timber framed between the purlins, or iron rods running from the head of one truss to the foot of the next.

21. Hip rafters, being deeper than the common rafters, are visible inside when the roof is ceiled, and should be covered with a casing.

22. Hips should stand perfectly at an angle of 45 degrees with the plates on plan, as by this arrangement the rafters on either side are equal in length, inclination, and bevel at the ends, making the construction both symmetrical and economical.

23. When the span is of such extent that the end purlin is longer than those of the side bays, a half truss should be introduced at the center of the end, with its tie-beam trimmed into the end transverse truss.

24. All the abutment joints in a framed truss should be at right angles with the direction of thrust, and when this is parallel with the edges of the member, the shoulders may be cut square with the back of such member.

25. To resist the racking movement in roofs, an effectual plan consists in the employment of wind ties of

iron. These extend usually from the head of one principal to the foot of the next principal, but one on the same side of the roof, and again from the head of this latter principal to the foot of the first one, so that the tie rods cross one another in the form of an X. It is difficult to estimate the stress which will come upon these ties; but very small sections, say from 5/8 in. to 3/4 in., will generally suffice for the purpose.

26. The amount of horizontal thrust at the foot of a principal rafter depends partly upon the weight of the truss and the loads or stresses which it has to sustain, and partly upon the inclination of the rafter. The lower the pitch of the roof, the greater is the proportion of thrust to weight, so that for roofs flatter than quarter pitch stronger tie-beams will be necessary.

27. In queen post trusses the position of the queen posts may vary. Generally, however, when there are no rooms in the roof, they are placed at one-third of the span from the wall.

28. When rooms are formed in queen post roofs, the distance between the queens may conveniently be half the span or more, but in such instances the depth of tie-beam should be increased.

29. The best form of roof truss to be used in any situation may be determined by the following considerations: (1) The parts of the truss between the points of support should not be so long as to have any tendency to bend under the thrust — therefore, the lengths of the parts under compression should not exceed twenty times their smallest dimensions; (2) The distance apart of the purlins should not be so great as to necessitate the use of either purlins or rafters too large for convenience or economy.

30. It has been found by experience that these objects

can be attained by limiting the distance between the points of support on the principal rafter to 8 ft., and upon the tie-beam to 15 ft.

31. To determine the form of roof truss for any given span, it is, therefore necessary first to decide the pitch, then roughly to draw the principal rafters in position, ascertain their length, divide them into portions 8 ft. long, and place a strut under each point of division. By this it will be seen that a king post truss is adapted for a roof, with principal rafters 16 ft. long—i. e., those having a span of 30 ft.

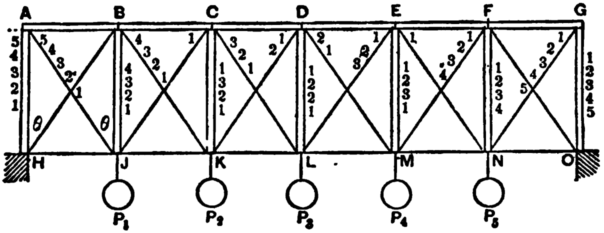


FIG. 50.

32. A queen post truss would be adapted to a roof with principals 24 ft. long—i. e., about 45 ft. span. For greater spans, with longer principals, compound roofs are required.

Pratt or Whipple Truss.—Referring to Fig. 50, the diagonals in this truss are ties, and the verticals are struts or columns.

Calculation by the Method of Distribution of Strains.—Consider first the load P. The truss having 6 bags or panels 5-6 of the load is transmitted to the abutment H, and 1-6 to the abutment O, on the principle of the lever.

As the five-sixths must be transmitted through JA, and AH, write on these members the figure 5. The one-sixth is transmitted successively through JC, CK, KD, DL, etc., passing alternately through a tie, and a strut. Write on these members up to the strut GO inclusive the figure 1. Then consider the load P_2, of which four-sixths goes to AH, and two-sixths to GO. Write on KB, BJ, JA, and AH, the figure 4, and on KD, DL, LE, etc., the figure 2. The load P_3 transmits 3/6 in each direction; write 3 on each of the members through which this stress passes, and so on for all the loads, when the figures on the several members will appear as on the cut. Adding them up, we have the following totals:

Tension on diagonals

AJ	BH	BK	CJ	CL	DK	DM	EL	EN	FM	FO	GN
15	0	10	1	6	3	3	3	6	1	10	0	15

Compression on verticals

AH	*BJ*	*CK*	*DL*	*EM*	*FN*	*GO*
15	10	7	6	7	10	15

Each of the figures in the first line is to be multiplied by 1/6 P × secant of angle HAJ, or 1/6 P × AJ ÷ AH, to obtain the tension, and each figure in the lower line is to be multiplied by 1/6 P to obtain the compression. The diagonals HB and FO receive no strain.

It is common to build this truss with a diagonal strut at HB instead of the post HA and the diagonal AJ; in which case 5/6 of the load P is carried through JB and the strut BH, which latter then receives a strain = 15/6 P × secant of HBJ.

The strains in the upper and lower horizontal members or chords increase from the ends to the center, as shown in the case of the Burr truss. AB receives a thrust equal to the horizontal component of the tension in AJ, or 15/6 P × tan AJB. BC receives the same thrust + the horizontal component of the tension in BK, and so on. The tension in the lower chord of each panel is the same as the thrust in the upper chord of the same panel. (For calculation of the chord strains by the method of moments, see below.)

The maximum thrust or tension is at the center of the chords and is equal to $\frac{WL}{8D}$, in which W is the total load supported by the truss, L is the length, and D the depth. This is the formula for maximum stress in the chords of a truss of any form whatever.

The above calculation is based on the assumption that all the loads P_1, P_2, etc., are equal. If they are unequal the value of each has to be taken into account in distributing the strains. Thus the tension in AJ, with unequal loads, instead of being $15 \times 1/6\,P$ secant θ would be sec. $\theta \times (5/6P_1 + 4/6P_2 + 3/6P_3 + 2/6P_4 + 1/6P_5)$. Each panel load, P_1, etc., includes its fraction of the weight of the truss.

General Formula for Strains in Diagonals and Verticals.—Let n = total number of panels, x = number of any vertical considered from the nearest end, counting the end as 1, r = rolling load for each panel, P = total load for each panel.

$$\text{Strain on verticals} = \frac{[(n-x)+(n-x)^2-(x-1)+(x-1)^2]P}{2n} + \frac{r(x-1)+(x-1)^2.}{2n}$$

For a uniformly distributed load, leave out the last term.

$$[r(x-1) + (x-1)^2] \div 2n.$$

Strain on principal diagonals = strain on verticals × secant θ, that is secant of the angle the diagonal makes with the vertical.

Strain on the counterbraces: The strain on the counterbrace in the first panel is 0, if the load is uniform.

On the 2d, 3d, 4th, etc., it is P secant $\times \frac{1}{n}$, $\frac{1+2}{n}$, $\frac{1+2+3}{n}$, etc., P being the total load in one panel.

Strain in the Chords—Method of Moments.—Let the truss be uniformly loaded, the total load acting on it = W. Weight supported at each end, or reaction of the abutment = W/2. Length of the truss = L. Weight on a unit of length = W/L. Horizontal distance from the nearest abutment to the point (say M in Fig. 50) in the chord where the strain is to be determined = x. Horizontal strain at that point (tension on the lower chord, compression in the upper) = H. Depth of the truss = D. By the method of moments we take the difference of the moments, about the point M, of the reaction of the abutment and of the load between M and the abutments, and equate that difference with the moment of the resistance, or of the strain in the horizontal chord, considered with reference to a point in the opposite chord, about which the truss would turn if the first chord were severed at M.

The moment of the reaction of the abutment is Wx/2. The moment of the load from the abutment to M is W/Lx × the distance of its centre of gravity from M, which is x/2, or moment = $Wx^2 \div 2L$. Moment of the stress in the chord$=HD=\frac{Wx}{2}-\frac{Wx^2}{2L}$, whence $H=\frac{W}{2D}(x-\frac{x^2}{L})$. If $x=0$ or L, $H=0$. If $x=L/2$, $H=\frac{WL}{8D}$, which is the horizontal strain at the middle of the chords, as before given.

The Howe Truss.—(*Fig.* 51.) In the Howe truss the diagonals are struts, and the verticals are ties. The calculation of strains may be made by the same method as that already described for the Pratt truss.

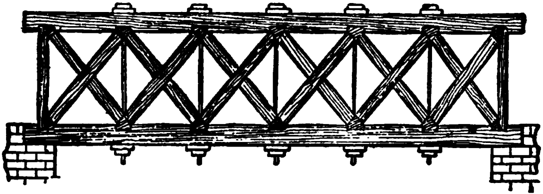


FIG. 51.

In the illustration shown in Fig. 52 is represented an ordinary lattice bridge which may have any ordinary span from 50 to 125 feet. No. 8 is the elevation of the common lattice bridge; No. 9, a section of the same when the roadway is above the latticed sides; and No. 10, a section when the roadway is supported on the under side of the lattice. No. 11, plan of one of the latticed sides.

Fig. 53 exhibits an elevation and details for an improved "Steele" lattice and trussed bridge, which is intended for long spans. The example shown was built over a span of more than 200 feet. The arch shown in

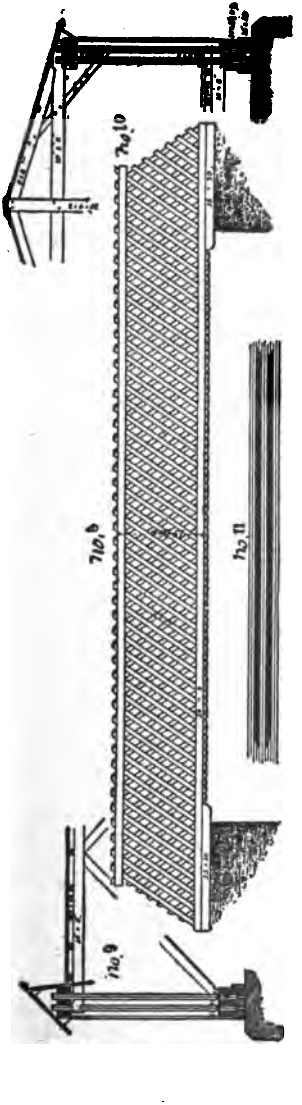


FIG. 52.

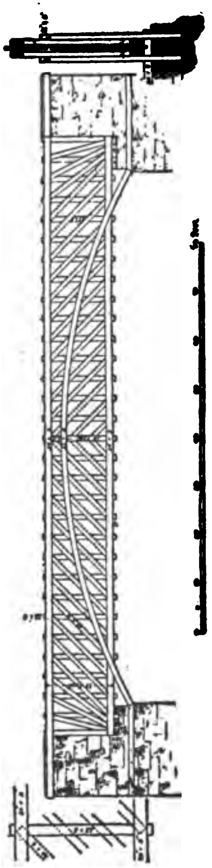


FIG. 53.

the work adds to the stability of the work very materially.

The details shown are self-evident and hardly require explanation.

In building a Howe truss, it is quite essential that the chords be arched or cambered. There are several ways of getting this camber, but the one recommended by Prof. De Volson Wood of Stevens' Institute, Hoboken, N. J., is perhaps, about the best. He says: "Camber may be accomplished in various ways. Having determined the length of the main braces for straight chords, if their length be slightly increased, beginning with nothing in the center and increasing gradually towards the ends, any desired camber may be secured. This will give an arch form." The result, in an exaggerated form, is shown in Fig. 53a. The bolts shown are all supposed to radiate to the center of the arch.

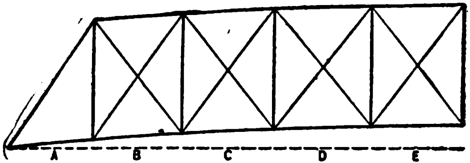


Diagram in Which the Panels at A is Shortest, and at E Longest. The Bolts Radiate to the Center of the Arch.

FIG. 53a.

The object of cambering a truss is to allow for any settlement which may occur after completion, and also to prevent the truss from deflecting below a horizontal line when taxed to its maximum capacity. Some engineers allow 1-inch camber for a span of 50 feet; 2 inches for 100 feet, etc., while some, depending on the accuracy of their work, allow only one-half this amount. By cambering the horizontal timbers it is manifest that they must be made longer than the straight line which joins their ends. The increase in length of the lower chords due to cambering would be so trifling that in ordinary practice it could be entirely disregarded. Not so, however, with the upper chord; the increase in length of this member would be quite an appreciable quantity, because the top chord is cambered to a curve which is concentric to the curve of the lower one.

Trautwine and other authorities give a rule for determining this increase when the depth, the camber, and the span are given, providing, however, that the camber does not exceed one-fiftieth of the span,

$$\text{Increase} = \frac{\text{depth} \times \text{camber} \times 8}{\text{span}}$$

using either feet or inches in the calculations. By cambering the truss the distance between the suspension rods on the upper chords will necessarily be greater than the distance between the rods on the lower chords. The panels are not strictly parallelograms, the rods converging somewhat. By dividing the total increase in length of the upper chord by the number of panels in the truss we obtain the increase per panel. This, of course, will effect the length of the braces, and great care should be taken to cut these to the proper length. Trautwine also gives a method for finding the length of the braces in cambered trusses, but while the method shown is practically correct, in so far as lines are concerned, yet it could not be applied very well in a timber truss, at least, not so well as the method shown previously.

It must be remembered, that in calculating strains in trusses, skeleton diagrams are used, and the lines composing these diagrams are generally taken or drawn through the axes of the various members. These lines usually meet at a common point of intersection as will be seen from the dotted lines in Fig. 53b. But in practice these lines do not always thus meet. The method shown by Trautwine is that of finding the length of the hypothenuse AC of the right angled triangle ABC; and even were these axial lines to meet at a common point of intersection the rule would not apply on account of the angle blocks taking up part of the distance. The best way to get the length would be to lay out one panel full size.

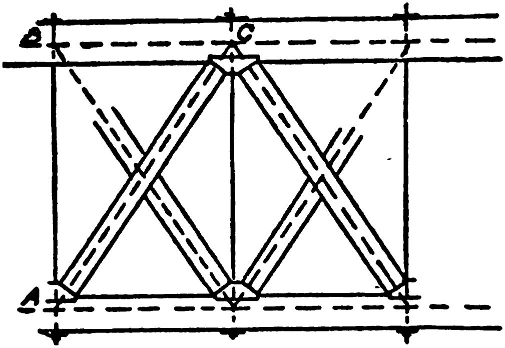


FIG. 53b.

Another branch of timber framing is that of "shoring and needling," which may be analyzed as follows:

A system of raking shores, consists of from one to four inclined timbers ranged vertically over each other, their lower ends springing from a stout sole-piece bedded in the ground, and their upper ends abutting partly against a vertical plank secured to the face of the wall and partly against the "needles"—horizontal projections that penetrate the wall-plate and the wall for a short distance.

The needles are generally cut out of 3-inch by 4½-inch stuff, the entering end reduced to 3-inch by 3-inch for convenience in entering an aperture formed by removing a header from the wall. The shouldered side is placed upwards, and cleats are fixed above them into the wall-plate to strengthen their resistance to the sliding tendency of the shore. They are preferably sunk into the plate at the top end.

The head of the raker should be notched slightly over the needle, to prevent its being knocked aside, or moving out of position in the event of the wall settling back.

The top shore in a system is frequently made in two lengths for convenience of handling, and the upper one is known as the "rider," the supporting shore being termed the "back shore."

The rider is usually set up to its bearing with a pair of folding wedges introduced between the ends of the two shores.

STEEL CONSTRUCTION.

BEAMS AND GIRDERS.

A Beam is any part of a machine or structure which is loaded in any way that tends to bend it, but usually transversely. A Girder is a large or main beam supported at both ends. A Cantilever is a beam unsupported at one end and fixed at the other. A Continuous Beam has more than two supports.

Beams of timber, cast iron, and wrought iron are used; but the best and most usual material is rolled steel made by the Bessemer, or the Siemens-Martin open-hearth acid process. Such steel breaks under a tension of 28 to 32 tons per square inch, elongates about 20 per cent in a length of 8 in., and contracts 40 per cent in area.

The main beams of bridges are styled Trusses, the smaller cross beams Joists. Rolled joists are of uniform I section from end to end. They are much used in large buildings and other structural work, also for light crane girders; being cheap, convenient, and readily obtainable. But the use of rolled joists for long spans involves some waste of material, owing to the section being uniform, instead of proportional to the varying bending moment.

TABLE 5

SAFE DISTRIBUTED LOADS ON HEAVY STEEL JOISTS.

(The Factor of Safety is 5.)

Depth.	Width.	Mean Thickness		Weight per Foot.	Clear Span.					
		Flange.	Web.		10 ft.	12 ft.	16 ft.	20 ft.	24 ft.	28 ft.
in.	in.	in.	in.	lb.	ton.	ton.	ton.	ton.	ton.	ton.
20	7.5	1.00	0.60	89	..	..	43	35	29	25
18	7	0.94	0.55	75	..	..	34	27	22	19
16	6	0.85	0.56	62	..	32	24	19	16	13
15	6	0.85	0.54	59	35	29	22	17	14	..
14	6	0.85	0.51	57	31	26	19	15	..	..
12	6	0.87	0.51	54	26	22	16	13	..	..
12	5	0.65	0.44	39	18	15	11	9	..	..
10	5	0.60	0.48	35	13	11	8	..	..	..

Note.—The safe loads stated include in each case the weight of the joist itself.

Formula for safe load on a rolled steel joist in terms of its weight:

$$W = 0.7\ (w—0.3BH)\ \frac{H}{L}.$$

In which

W = safe distributed load (tons).

W = weight joist per lineal foot (pounds).

B = breadth of flange (inches).

H = total depth of joist (inches).

L = clear span (feet).

This formula gives safe loads slightly less than those given in table 5.

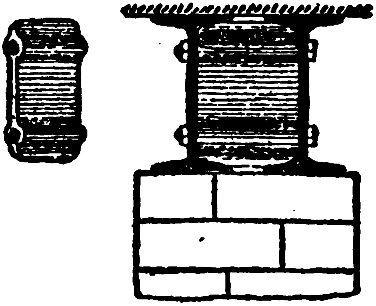


FIG. 54.

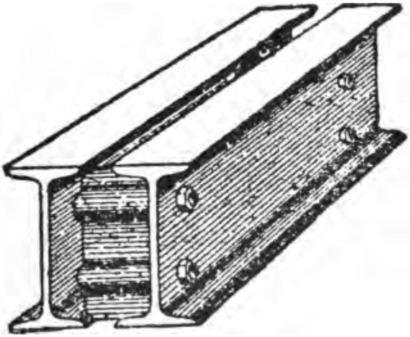


FIG. 55.

Compound Joists.—Figs. 54 and 55 are made by connecting together two or three simple joists by means of cast iron distance pieces and bolts. These answer well for the beams over wide doorways and windows.

A Compound, or Box Girder.—Fig. 56 is built up of either rolled joists or channels; and two or more plates. Such a girder can rapidly be put together from stock sections.

A Plate Girder.—Fig. 57 has top and bottom flanges, or booms, of two or more plates connected by angle irons to a plate web, stiffened by T-irons. A double-webbed box girder is less liable to lateral buckling.

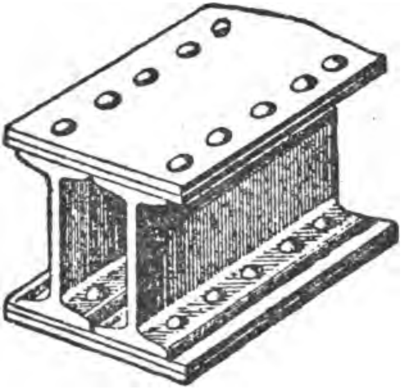


FIG. 56.

A Lattice Girder.—Fig. 59 has its flanges connected by a system of diagonal bars or trellis work, usually made of channel bars riveted together.

A Warren Girder is the commonest example of a braced girder, of which there are many types. Its flanges are connected by substantial bars inclined at 60°, the joints being made with pins. Those bars in tension are termed Ties, and those in compression struts.

A Bowstring Girder has a stiff arch above, and from it is hung the "platform" which ties the ends of the arch together.

The Reactions are the passive upward supporting forces induced by the applied forces or loads.

A Concentrated or Local Load (W tons) is applied at a single section, while a uniformly distributed load (W tons per foot run) is spread equally all over the beam.

A Dead Load is gradually applied, and remains steady, or constant in amount.

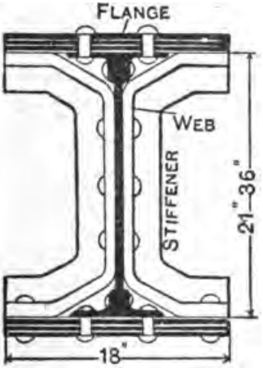


FIG. 57.

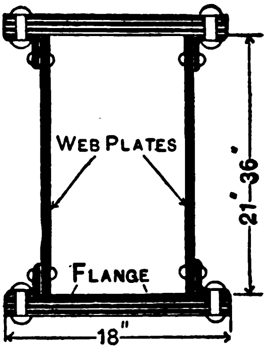


FIG. 58.

DIMENSIONS GIVEN REFER TO STANDARD SECTIONS.

A Live Load comes on suddenly, and causes vibration. A live load is twice as destructive as a dead load of the same nominal amount. Hence a live load is reduced to an equivalent dead load by doubling it.

The Load on a Floor is reckoned in pounds per sq. ft. It consists of the weight of the materials composing the floor (dead load), and the weight of the persons, goods or machinery (live load). The dead load is usually reckoned at from 70 to 80 lbs. per sq. ft.

The following table gives the total loads (live and dead) that may be allowed for in designing various kinds of floors. As the loads are *mixed,* a suitable factor of safety, for steel joists, is *four.*

Nature of Floor.	Load. lb. per sq. ft.
For houses or offices	150
For public halls and schools	180
For warehouses	350
Floors carrying heavy machinery	500

The *Breaking Load* of a beam is that *dead* load which just suffices to cause fracture. The *Working* or *Safe Load* is the greatest *dead* load which the beam is permitted to carry, or its equivalent *live* load. The latter must always be so small as to put all danger of rupture out of the question. Hence the need for a factor of safety.

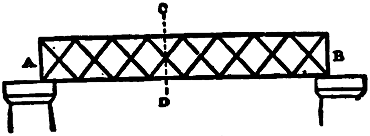


FIG. 59.

The *Clear Span* of a beam is the distance between the vertical faces of the supports or *Abutments.* The *Effective Span* (L inches or feet) is the distance between the centers of the bearing surfaces on which the beam rests.

The *Effective Depth* of a girder is the distance between the centers of area of the flanges. It is the length of the resistance arm (in inches or feet).

The *Vertical Shearing Force* (S tons) at a section of a beam is the algebraic sum of all the external forces acting on *either* side of it. At every section the *Horizontal Shearing Force* is equal to the vertical shearing force.

The *Bending Moment* or *Moment of Flexure* (M inch-tons) at a cross section of a beam is the algebraic sum of the moments of all the external forces on *either* side of that section. Its value depends on the position and magnitude of the loads.

The *Working Moment of Resistance* (M inch-tons) of a section of a beam measures the local ability of the beam to resist, without distress, the bending moment actually applied; to which it is numerically equal, though of opposite sign. When speaking, without qualification, of the moment of resistance, the *greatest* safe working value is meant.

The *Ultimate Moment of Resistance* is numerically equal to the moment of the bending couple which will just break the beam. It is not possible to calculate this in any rational manner; but an empirical rule is given by Sir B. Baker, which may be put into mathematical form as follows:

Baker's Rule.—Let M inch-tons be the moment of resistance calculated by the ordinary methods; using, however, the *ultimate* tensile stress on the material in place of the safe stress. Let a square inches be the area of the cross section of the beam, and A square inches the area of the enclosing rectangle. Then the *ultimate* moment of resistance is

$$M + 70 \text{ per cent of } M \times \frac{a}{A} = M\left(1 + 0.7\frac{a}{A}\right)$$

from which the load required to *break* a beam of given span may be approximately calculated.

In a *Beam of Uniform Strength* the greatest safe or working moment of resistance of each cross section equals the bending moment at that section. Such beams are commonly "fishbellied," as crane girders; or "saddle-backed," as bridge-girders; or have flanges of varying thickness.

Stress (f tons per square inch) is the resistance of the material per unit area of cross section induced by the load; or, more generally, any pair of equal and directly opposed forces (P tons).

Strain is the fractional change of length per unit length caused by the load; or, more generally, any alteration of shape or size due to stress.

The *Elastic Limits* of a material are those particular stresses, of tension and pressure, at which the ratio of stress to strain first ceases to be constant, or at which "permanent set" begins. Strength and deflection formulæ apply only within these limits of stress.

The *Modulus of Elasticity* (E tons per square inch) of a material is the ratio of stress to strain within the elastic limits. E is about 13,400 for mild steel, but may be taken as 9,000 for *riveted* work.

The *Neutral Surface* of a beam is that layer of fibers, or of particles, which is neither compressed nor extended when the beam is bent. The *Neutral Axis* of a section is the line in which this section cuts the neutral surface. It passes through the center of area or *Centroid* of the section, provided the modulus of elasticity of the material in tension equals that in compression. The stress on any part of a section varies as its distance from the neutral axis.

Camber is the initial rise in the center given to a girder in order to avoid any appearance of sagging or deflection below the horizontal. An average value is 3/8-in. per 10 ft. of clear span.

The *Modulus of a Section* (Z inch^3) of a beam is a geometrical quantity whose value depends solely on the

TABLE 6.

SAFE LOADS UNIFORMLY DISTRIBUTED FOR STANDARD AND SPECIAL I-BEAMS

IN TONS OF 2,000 LBS.

Distance between Supports in Feet	24" I	Add for every lb. increase in weight	20" I		Add for every lb increase in weight	18" I	Add for every lb. increase in weight	15" I			Add for every lb. increase in weight
	80 lbs.		80 lbs.	65 lbs.		55 lbs.		80 lbs.	60 lbs.	42 lbs.	
12	77 33	53	65 18	51 98	44	39 29	39	47 14	36 09	26 18	.33
13	71 38	48	60 16	47 98	40	36 27	36	43 51	33 31	24 17	.30
14	66.28	45	55 87	44 56	37	33 68	34	40 40	30 93	22.44	28
15	61 86	42	52 14	41 59	35	31 43	31	37 71	28 87	20 94	.26
16	58 00	39	48 88	38 99	33	29 47	29	35 35	27.07	19 63	24
17	54 58	37	46 01	36 69	31	27 74	28	33 27	25.47	18 48	.23
18	51 56	35	43 45	34.66	29	26 19	26	31 42	24 06	17 45	.22
19	48 84	33	41 17	32.83	28	24.82	25	29 77	22 79	16.53	.21
20	46 40	32	39 11	31 19	26	23 58	24	28 28	21.65	15 71	.20
21	44 19	30	37 24	29 70	25	22 45	22	26 94	20 62	14.96	19
22	42 18	29	35.55	28 35	24	21 43	21	25.71	19 68	14 28	18
23	40 35	27	34 01	27 12	23	20 50	20	24 59	18 83	13 66	17
24	38 67	26	32 59	25 99	22	19 65	20	23.57	18 04	13.09	.16
25	37 12	25	31 29	24 95	21	18 86	19	22 63	17 32	12 57	16
26	35 69	24	30 08	23 99	20	18 14	18	21 76	16 66	12 08	.15
27	34 37	23	28 97	23 10	19	17 46	17	20 95	16 04	11 64	14
28	33.14	23	27 93	22 28	19	16 84	17	20 20	15 47	11.22	14
29	32 00	22	26 97	21 51	18	16 26	16	19 51	14 93	10 83	13
30	30.93	21	26 07	20 79	17	15 72	16	18 86	14 43	10.47	.13
31	29.94	20	25 23	20 12	17	15 21	15	18 25	13.97	10 13	13
32	29.00	20	24 44	19 49	16	14 73	15	17 68	13.53	9 82	.12
33	28.12	19	23 70	18 90	16	14 29	14	17 14	13 12	9 52	12
34	27.29	19	23 00	18 35	15	13 87	14	16.64	12.74	9.24	.11
35	26.51	18	22 35	17 82	15	13 47	13	16.16	12.37	8.98	.11
36	25.78	.18	21 73	17 33	15	13 10	13	15.71	12.03	8.73	.11

Safe loads given include weight of beam. Maximum fiber stress, 16,000 lbs. per square inch.

TABLE 6—Continued.

SAFE LOADS UNIFORMLY DISTRIBUTED FOR STANDARD AND SPECIAL I-BEAMS

IN TONS OF 2000 LBS.

Distance between Supports in Feet	12" I		Add for every lb. increase in weight	10" I	Add for every lb. increase in weight	9" I	Add for every lb. increase in weight	Distance between Supports in Feet	8" I	Add for every lb. increase in weight
	40 lbs.	31.5 lbs		25 lbs.		21 lbs.			18 lbs.	
12	19.92	15.99	.26	10.85	22	8.39	.20	5	15.17	.42
13	18.39	14.76	.24	10.02	.20	7.74	.18	6	12.64	.35
14	17.08	13.70	.23	9.30	.19	7.19	.17	7	10.84	.30
15	15.94	12.79	.21	8.68	.17	6.71	.16	8	9.48	.26
16	14.94	11.99	.20	8.14	16	6.29	.15	9	8.43	.23
17	14.06	11.29	.19	7.66	.15	5.92	.14	10	7.59	.21
18	13.28	10.66	.18	7.24	14	5.59	.13	11	6.90	.19
19	12.58	10.10	.17	6.86	14	5.30	.12	12	6.32	.18
20	11 95	9 59	.16	6.51	.13	5.03	.12	13	5.83	.16
21	11.38	9.14	.15	6.20	.12	4.79	.11	14	5.42	.15
22	10.87	8.72	.14	5.92	.12	4.58	.11	15	5.06	.14
23	10.39	8.34	.14	5.66	.11	4.38	.10	16	4.74	.13
24	9.96	7.99	.13	5.43	.11	4.19	.10	17	4.46	.12
25	9.56	7.67	.13	5.21	.10	4.03	.09	18	4.21	.12
26	9.19	7 38	.12	5.01	.10	3.87	.09	19	3.99	.11
27	8.85	7.11	.12	4.82	.10	3.73	.09	20	3.79	.11
28	8.54	6.85	.11	4.65	.09	3.59	.08	21	3.61	.10
29	8.24	6.62	.11	4.49	.09	3.47	.08	....	......	....
30	7 97	6.40	.11	4.34	.09	3.36	.08	....	......	....

Safe loads given include weight of beam. Maximum fiber stress, 16,000 lbs. per square inch.

TABLE 6—Continued.

SAFE LOADS UNIFORMLY DISTRIBUTED FOR STANDARD AND SPECIAL I-BEAMS

IN TONS OF 2,000 LBS.

Distance between Supports in Feet	7" I	Add for every lb. increase in weight	6" I	Add for every lb. increase in weight	5" I	Add for every lb. increase in weight	4" I	Add for every lb. increase in weight	3" I	Add for every lb. increase in weight
	15 lbs.		12.25 lbs.		9.75 lbs.		7.5 lbs.		5.5 lbs.	
5	11.04	.36	7.75	.31	5.16	.26	3.18	.21	1.76	.16
6	9.20	.30	6.46	.26	4.30	.22	2.65	.18	1.47	.13
7	7.89	.26	5.54	.22	3.69	.19	2.27	.15	1.26	.11
8	6.90	.23	4.84	.19	3.23	.16	1.99	.13	1.10	.10
9	6.13	.20	4.31	.17	2.87	.14	1.77	.12	0.98	.09
10	5.52	.18	3.88	.16	2.58	.13	1.59	.11	0.88	.08
11	5.02	.16	3.52	.14	2.35	.12	1.45	.10	0.80	.07
12	4.60	.15	3.23	.13	2.15	.11	1.33	.09	0.73	.07
13	4.25	.14	2.98	.12	1.98	.10	1.22	.08	0.68	.06
14	3.94	.13	2.77	.11	1.84	.09	1.14	.08	0.63	.06
15	3.68	.12	2.58	.10	1.72	.09	1.06	.07	0.59	.05
16	3.45	.11	2.42	.10	1.61	.08	0.99	.07	0.55	.05
17	3.25	.11	2.28	.09	1.52	.08	0.94	.06	0.52	.05
18	3.07	.10	2.15	.09	1.43	.07	0.88	.06	0.49	.04
19	2.91	.09	2.04	.08	1.36	.07	0.84	.06	0.46	.04
20	2.76	.09	1.94	.08	1.29	.07	0.80	.05	0.44	.04
21	2.63	.09	1.85	.07	1.23	.06	0.76	.05	0.42	.04

Safe loads given include weight of beam. Maximum fiber stress 16,000 lbs. per square inch.

TABLE 6—Continued.

SAFE LOADS UNIFORMLY DISTRIBUTED FOR STANDARD AND SPECIAL CHANNELS

IN TONS OF 2,000 LBS.

Distance between Supports in Feet	15" ⊏	Add for every lb. increase in weight	12" ⊏	Add for every lb. increase in weight	10" ⊏	Add for every lb. increase in weight	9" ⊏	Add for every lb. increase in weight
	33 lbs.		20.5 lbs.		15 lbs.		13.25 lbs.	
10	22.23	.39	11.39	.32	7.14	.26	5.61	.24
11	20.20	.35	10.35	.29	6.49	.24	5.10	.21
12	18.52	.33	9.49	.26	5.95	.22	4.68	.20
13	17.10	.30	8.76	.24	5.49	.20	4.32	.18
14	15.87	.28	8.14	.23	5.10	.19	4 01	.17
15	14.82	.26	7.59	.21	4.76	.17	3.74	.16
16	13.89	.24	7.12	.20	4.46	.16	3.51	.15
17	13.07	.23	6.70	.18	4.20	.15	3.30	.14
18	12.35	.22	6.33	.18	3.96	.14	3.12	.13
19	11.70	.21	5.99	.17	3.76	.14	2.95	.12
20	11.11	.20	5.70	.16	3.57	.13	2.81	.12
21	10.58	.19	5.42	.15	3.40	.12	2.67	.11
22	10.10	.18	5.18	.14	3.24	.12	2.55	.11
23	9.66	.17	4.95	.14	3.10	.11	2.44	.10
24	9.26	.16	4.75	.13	2.97	.11	2.34	.10
25	8.89	.16	4.56	.13	2.85	.10	2.24	.09
26	8.55	.15	4.38	.12	2.74	.10	2.16	.09
27	8.23	.14	4.22	.12	2.64	.10	2.08	.09
28	7.94	.14	4.07	.11	2.55	.09	2.00	.08
29	7.66	.13	3.93	.11	2.46	.09	1.93	.08
30	7.41	.13	3.80	.11	2.38	.09	1.87	.08

Safe loads given include weight of channel. Maximum fiber stress, 16,000 lbs. per square inch.

shape and size of the section. When multiplied by the safe limiting stress, it gives the moment of resistance of the section.

The *Moment of Inertia of a Section* (I inch^4) relative to the neutral axis is a quantity which, when divided by the distance of the extreme fibers from that axis, gives the modulus of the section.

To find the moment of inertia of a plane figure about a given axis, multiply the area of a small part of the figure by the square of its distance from that axis, and repeat the operation several times till the whole figure has been dealt with piecemeal. The sum of all these products is the quantity sought.

Table 6 gives the safe loads for I beams ranging in width from 3 inches to 24 inches; and with supports at various distances apart. These loads include the weight of the beam, and are assumed to be uniformly distributed over its entire length. In order to ascertain the net load that the beam will carry, deduct weight of beam. The weight in pounds per foot is given at head of columns.

Table 7 gives safe loads evenly distributed for channel bars ranging in width from 3 inches to 15 inches. Weight of bar pounds per foot at top of columns.

TABLE 7.

SAFE LOADS UNIFORMLY DISTRIBUTED FOR STANDARD AND SPECIAL CHANNELS.

IN TONS OF 2,000 LBS.

Distance between Supports in Feet	8″ ⊏	Add for every lb. increase in weight	7″ ⊏	Add for every lb. increase in weight	6″ ⊏	Add for every lb. increase in weight	5″ ⊏	Add for every lb. increase in weight	4″ ⊏	Add for every lb. increase in weight	3″ ⊏	Add for every lb. increase in weight
	11.25 lbs.		9.75 lbs.		8 lbs.		6.5 lbs.		5.25 lbs.		4 lbs.	
5	8.61	.42	6 68	.36	4 62	.31	3 16	26	2.02	21	1 16	16
6	7.18	.35	5 57	.30	3 85	26	2 63	22	1.68	.18	.97	13
7	6.15	.30	4.77	.26	3.30	.22	2.26	.19	1.44	.15	.83	.11
8	5.38	.26	4.18	.23	2.89	.19	1 98	.16	1.26	.13	.73	.10
9	4 78	.23	3 71	.20	2 57	.17	1.76	.14	1.12	.12	.64	09
10	4.31	.21	3.34	.18	2 31	16	1.58	.13	1.01	.11	.58	08
11	3.91	.19	3.04	.16	2 10	14	1.44	.12	.92	.10	.53	.07
12	3.59	.18	2.78	.15	1 93	13	1.32	.11	.84	09	.48	07
13	3.31	.16	2.57	.14	1.78	.12	1 22	10	.78	.08	.45	.06
14	3.08	15	2.39	.13	1.65	.11	1 13	09	.72	.08	.41	.06
15	2 87	14	2.23	.12	1.54	10	1 05	09	.67	.07	.39	.05
16	2 69	.13	2 09	11	1 44	10	99	08	.63	.07	.36	.05
17	2 53	12	1 96	11	1 36	09	.93	08	.59	06	.34	.05
18	2 39	.11	1.86	.10	1.28	.09	.88	07	.56	.06	.32	04
19	2.27	.11	1.76	09	1 22	08	83	07	.53	.06	.31	.04
20	2.15	.11	1.67	.09	1 16	.08	79	07	.51	.05	.29	.04
21	2 05	10	1.59	09	1 10	07	75	06	.48	05	.28	.04
22	1.96	.10	1 52	08	1 05	07	72	06	.46	.05	.26	.04
23	1.87	.09	1 45	.08	1 00	07	.69	06	.44	.05	.25	.03
24	1.79	09	1 39	08	.96	06	.66	05	.42	.04	.24	.03
25	1 72	.08	1 34	07	.92	06	63	05	.40	.04	.23	.03

Safe loads given include weight of channel. Maximum fiber stress, 16,000 lbs. per square inch.

TABLE 8.

SPACING OF STANDARD I-BEAMS FOR UNIFORM LOAD OF 100 LBS. PER SQUARE FOOT

PROPER DISTANCE IN FEET CENTER TO CENTER OF BEAMS

Distance between Supports in Feet	24" I	20" I		18" I	15" I			12" I		10" I
	80 lbs.	80 lbs.	65 lbs.	55 lbs.	80 lbs.	60 lbs.	42 lbs.	40 lbs.	31.5 lbs.	25 lbs.
12	128.9	108 6	86 6	65 5	78.6	60.1	43.6	33.2	26 6	18.1
13	109.8	92.6	73.8	55.8	67.0	51.3	37.2	28.3	22.7	15.4
14	94.7	79.8	63.7	48.1	57.7	44.2	32 1	24.4	19.6	13.3
15	82.5	69.5	55.5	41.9	50.3	38.5	27.9	21.3	17.1	11 6
16	72.5	61.1	48.7	36.8	44.2	33.8	24.5	18.7	15.0	10.2
17	64.2	54.1	43.2	32.6	39.2	30.0	21.7	16 5	13.3	9 0
18	57.3	48.3	38.5	29.1	34.9	26.7	19.4	14.8	11 8	8.0
19	51.4	43.3	34.6	26 1	31.3	24.0	17.4	13 2	10 6	7 2
20	46.4	39.1	31.2	23.6	28 3	21.7	15.7	12 0	9.6	6 5
21	42.1	35.5	28.3	21 4	25.7	19.6	14.2	10.8	8.7	5 9
22	38.4	32.3	25.8	19 5	23.4	17.9	13.0	9 9	7.9	5 4
23	35.1	29.6	23.6	17.8	21.4	16.4	11.9	9.0	7.3	4 9
24	32.2	27.2	21.7	16.4	19.6	15.0	10.9	8.3	6 7	4 5
25	29.7	25.0	20.0	15.1	18.1	13.9	10 1	7 7	6.1	4 2
26	27 5	23.1	18.5	13.9	16 7	12 8	9 3	7 1	5.7	3 9
27	25.5	21 5	17.1	12.9	15 5	11 9	8 6	6 6	5.3	3.6
28	23.7	20.0	15.9	12.0	14.4	11.0	8 0	6 1	4 9	3.3
29	22.1	18.6	14.8	11.2	13.5	10 3	7 5	5.7	4 6	3 1
30	20.6	17.4	13.9	10 5	12.6	9.6	7 0	5 3	4 3	2 9

For load of 200 lbs. per square foot divide the spacing given by 2. Maximum fiber stress, 16,000 lbs. per square inch.

TABLE 8—Continued.

SPACING OF STANDARD I-BEAMS FOR UNIFORM LOAD OF 100 LBS. PER SQUARE FOOT

PROPER DISTANCE IN FEET CENTER TO CENTER OF BEAMS

Distance between Supports in Feet	9" I	8" I	7" I	6" I	5" I	4" I	3" I
	21 lbs.	18 lbs.	15 lbs.	12.25 lbs.	9.75 lbs.	7.5 lbs.	5.5 lbs.
5	80.5	60.7	44 2	31.0	20 6	12 7	7.0
6	55.9	42.1	30.7	21 5	14 3	8.8	4.9
7	41.1	31.0	22.5	15.8	10.5	6.5	3.6
8	31 5	23.7	17 3	12 1	8.1	5.0	2 8
9	24.9	18 7	13.6	9 6	6.4	3.9	2.2
10	20.1	15.2	11.1	7.8	5.2	3.2	1.8
11	16 6	12 5	9.1	6 4	4.3	2.6	1.5
12	14.0	10 5	7.7	5.4	3 6	2.2	1.2
13	11.9	9.0	6.5	4.6	3 1	1 9	1.0
14	10.3	7.7	5 6	4.0	2 6	1.6	0.9
15	9.0	6.7	4.9	3.4	2.3	1.4	
16	7.9	5.9	4.3	3.0	2 0	1.2	
17	7.0	5 3	3.8	2.7	1.8	1.1	
18	6.2	4 7	3.4	2.4	1 6	.98	
19	5.6	4.2	3.1	2.2	1.4		
20	5.0	3.8	2.8	1.9	1.3		
21	4.6	3 4	2.5	1.8	1.2		
22	3.8	3 1	2.3	1.6	1.1		

For load of 200 lbs. per square foot divide the spacing given by 2. Maximum fiber stress, 16,000 lbs. per square inch.

General Formulæ on the Flexure of Beams of any Cross-Section.

Let A = area of section in square inches,
l = length of span in inches,
W = load uniformly distributed in pounds,
M = bending moment in inch-pounds,
h = height of cross-section, out to out, in inches,
n = distance of center of gravity of section, from top or from bottom, in inches,
f = stress per square inch in extreme fibers of beam, either top or bottom, in pounds, according as n relates to distance from top or from bottom of section,
D = maximum deflection in inches,
I = moment of inertia of section neutral axis through center of gravity,
I″ = moment of inertia of section neutral axis parallel to above, but not through center of gravity,
d = distance between these neutral axes,
S = section modulus,
r = radius of gyration in inches,
E = modulus of elasticity for steel 29.000,000;

Then: $$S = \frac{I}{n}, \qquad r = \sqrt{\frac{I}{A}},$$

$$M = \frac{fI}{n} = fS,$$

$$f = \frac{Mn}{I} = \frac{M}{S},$$

$$W = \frac{8\,f\,I}{l\,n} = \frac{8\,f}{l}\,S,$$

$$f = \frac{W l n}{8 I} = \frac{W l}{8\,S},$$

$$I'' = I + Ad^2,$$

$D = \frac{5\,Wl^3}{384\,EI}$ for beam supported at both ends and uniformly loaded,

$D = \frac{Pl^3}{48\,EI}$ for beam supported at both ends and loaded with a single load P at middle,

$D = \frac{Wl^3}{8\,EI}$ for beam fixed at one end and unsupported at the other and uniformly loaded,

$D = \frac{Pl^3}{3\,EI}$ for beam fixed at one end and unsupported at other and loaded with a single load P at the latter end.

The box, or compound girder, has already been alluded to. Tables 9 and 10 give the safe carrying capacity of box girders made up of 10-inch and 12-inch I beams. The distance center to center of bearing varies from 10 feet to 39 feet.

The values given in Tables 9 and 10 are founded upon the moments of inertia of the various sections. Deduc-

TABLE 9.

BEAM BOX GIRDERS

SAFE LOADS IN TONS UNIFORMLY DISTRIBUTED

2—10 in. I-Beams and 2 Plates 12 in. × ½ in.

2 Plates
12×½

10 in.
I-Beams
25.0 lbs.
per Foot

Distance Center to Center of Bearing in Feet	Safe Load Uniformly Distributed (Including Weight of Girder) in Tons of 2,000 lbs.	Weight of Girder (Including Rivet Heads) in Tons of 2,000 lbs.	Increase in Safe Load for $\frac{1}{16}$ in. Increase in Thickness of Flange Plates	Increase of Weight of Girder for $\frac{1}{16}$ in. Increase in Thickness of Flange Plates
10	45.00	0.47	3.06	0.02
11	40.92	0.52	2.78	0.03
12	37.50	0.56	2.55	0.03
13	34.62	0.61	2.35	0.03
14	32.15	0.66	2.18	0.03
15	30.00	0.70	2.04	0.04
16	28.13	0.75	1.90	0.04
17	26.47	0.80	1.80	0.04
18	25.00	0.84	1.70	0.04
19	23.69	0.89	1.60	0.05
20	22.50	0.94	1.53	0.05
21	21.43	0.99	1.45	0.05
22	20.46	1.03	1.38	0.05
23	19.57	1.08	1.32	0.06
24	18.75	1.13	1.28	0.06
25	18.00	1.17	1.22	0.06
26	17.31	1.22	1.18	0.06
27	16.67	1.27	1.13	0.07
28	16.07	1.31	1.10	0.07
29	15.52	1.36	1.05	0.07
30	15.00	1.41	1.02	0.07
31	14.52	1.45	0.99	0.08
32	14.07	1.50	0.96	0.08
33	13.64	1.55	0.92	0.08
34	13.23	1.59	0.90	0.08
35	12.85	1.64	0.88	0.09
36	12.50	1.69	0.85	0.09
37	12.16	1.74	0.83	0.09
38	11.84	1.78	0.81	0.09
39	11.54	1.83	0.78	0.10

Above values are based on maximum fiber stress of 15,000 lbs. per square inch; $\frac{13}{16}$ in. rivet holes in both flanges deducted. Weights of girders correspond to lengths center to center of bearings.

tions were made for the rivet holes in both flanges. The maximum stress in extreme fibers was limited to 15,000

TABLE 10.

BEAM BOX GIRDERS

SAFE LOADS IN TONS UNIFORMLY DISTRIBUTED

2—12 in. I-Beams and 2 Plates 14 in. × ½ in.

	2 Plates 14×½	12 in. I-Beams 40.0 lbs. per Foot		12 in. I-Beams 31.5 lbs. per Foot			2 Plates 14×½
Distance Center to Center of Bearings in Feet	Safe Load Uniformly Distributed (Including Weight of Girder) in Tons of 2,000 lbs.	Weight of Girder (Including Rivet Heads) in Tons of 2,000 lbs.	Increase in Safe Load for 1/16 in. Increase in Thickness of Flange Plates	Safe load Uniformly Distributed (Including Weight of Girder) in Tons of 2,000 lbs.	Weight of Girder (Including Rivet Heads) in Tons of 2,000 lbs.	Increase in Safe Load for 1/16 in. Increase in Thickness of Flange Plates	Increase in Weight of Girder for 1/16 in. Increase in Thickness of Flange Plates
10	73.33	0.65	4.33	65.97	0.57	4.40	0.03
11	66.67	0.72	3.93	59.98	0.63	3.99	0.03
12	61.11	0.78	3.61	54.98	0.68	3.66	0.03
13	56.40	0.85	3.33	50.75	0.74	3.38	0.04
14	52.37	0.91	3.09	47.12	0.80	3.14	0.04
15	48.89	0.98	2.88	43.98	0.85	2.93	0.04
16	45.83	1.05	2.70	41.23	0.91	2.75	0.05
17	43.13	1.11	2.55	38.80	0.97	2.58	0.05
18	40.73	1.18	2.40	36.65	1.02	2.45	0.05
19	38.60	1.24	2.27	34.72	1.08	2.31	0.05
20	36.67	1.31	2.17	32.99	1.14	2.19	0.06
21	34.92	1.37	2.07	31.42	1.19	2.09	0.06
22	33.33	1.44	1.97	29.99	1.25	2.00	0.06
23	31.88	1.50	1.88	28.68	1.31	1.92	0.07
24	30.55	1.57	1.80	27.48	1.36	1.83	0.07
25	29.33	1.63	1.73	26.39	1.42	1.75	0.07
26	28.20	1.70	1.66	25.37	1.48	1.68	0.08
27	27.16	1.76	1.60	24.44	1.53	1.63	0.08
28	26.19	1.83	1.55	23.56	1.59	1.57	0.08
29	25.28	1.89	1.49	22.75	1.65	1.51	0.08
30	24.44	1.96	1.44	21.99	1.70	1.47	0.09
31	23.65	2.02	1.40	21.28	1.76	1.42	0.09
32	22.92	2.09	1.35	20.62	1.81	1.37	0.09
33	22.22	2.16	1.32	20.00	1.87	1.33	0.10
34	21.57	2.22	1.27	19.41	1.93	1.29	0.10
35	20.95	2.29	1.23	18.85	1.99	1.26	0.10
36	20.37	2.35	1.20	18.32	2.05	1.22	0.10
37	19.82	2.42	1.17	17.83	2.10	1.19	0.11
38	19.29	2.48	1.14	17.37	2.16	1.15	0.11
39	18.81	2.55	1.11	16.92	2.21	1.13	0.11

Above values are based on maximum fiber stress of 15,000 lbs. per square inch; 13/16 in. rivet holes in both flanges deducted. Weights of girders correspond to lengths center to center of bearings.

lbs. per square inch, while in the tables on rolled steel beams a fiber stress of 16,000 lbs. was used. This reduction was made in order to amply compensate for the deterioration of the metal around the rivet holes from punching.

Box girders should not be used in damp or exposed places, since the interior surfaces do not readily admit of repainting.

Example.—A 13-in. brick wall 16 feet high is to be built over an opening of 25 ft. What will be the section of the girder required?

Answer.—Assuming 26 feet as the distance center to center of bearings the weight of the wall will be $26 \times 16 \times 121 = 50{,}336$ lbs., or 25.17 tons.

According to Table 10, a girder composed of two 12-in. beams each weighing 31.5 lbs. per foot and two $14 \times \frac{1}{2}$ in. flange plates will carry safely for a span of 26 ft. a uniformly distributed load of 25.37 tons, including its own weight. Deducting the latter, 1.48 tons given in the next column, we find 23.89 tons for the value of the safe net load, which is 1.28 tons less than required. From the following column we find that by increasing the thickness of the flange plates 1/16 in. we may add 1.68 tons to the allowable load. This will more than cover the difference. Hence, the required section will be two 12-in. beams 31.5 lbs. per foot and two $14 \times 9/16$ in. cover plates.

Notes on Steel and Iron.—1. The average weight of wrought iron is 480 lbs. per cubic foot. A bar 1 inch square and 3 feet long weighs, therefore, exactly 10 lbs. Hence:

To find the sectional area, given the weight per foot, multiply by 3/10.

To find the weight per foot, given the sectional area, multiply by 10/3.

2. The weight of steel is 2 per cent greater than that of wrought iron. To find sectional area, given weight per foot, divide by 3.4.

To find weight per foot, given sectional area, multiply by 3.4.

3. The center load at which a bar of wrought iron 1 in. square, and 12 in. center to center of points of support, will give way is very nearly one ton.

4. Within the elastic limit the extension and compression of wrought iron is very nearly $\frac{1}{10000}$ of its length for a stress of one ton per square inch.

For cast iron this ratio is $\frac{1}{5000}$ for tension, but becomes variable for compression.

5. The contraction or expansion of wrought iron under changes of temperature is about $\frac{1}{10000}$ of its length, for a variation of 15° Fahrenheit.

The stress thus induced, if the ends are held rigidly fixed, will be about one ton per square inch of cross-section.

6. The coefficient of expansion of wrought iron for 100° Fahrenheit is 0.000686. Therefore, for a variation in temperature of 125°, a bar of wrought iron 100 feet long will expand or contract 1.029 inches.

Conversely—A change in length of 1 inch per hundred feet would be produced by a variation in temperature of 121.5° Fahrenheit.

7. The melting point of iron and steel is about as follows: Wrought iron, 3,000° Fahrenheit; cast iron, 2,000° Fahrenheit; steel, 2,400° Fahrenheit.

8. The welding heat of wrought iron is 2,733° Fahrenheit.

Miscellaneous Notes.—1. Thrust of arch per lineal foot: $T = \frac{1.5\,wl^2}{r}$, in which w = load per square foot, r = rise in arch in inches, and l = span in feet.

2. Approximately the radius of gyration for a box section is 4/10 the least side.

Strength of Materials.—Table 11 gives the ultimate strength of the materials with which the millwright has to deal mainly:

TABLE 11.—ULTIMATE STRENGTH OF MATERIALS.

Material	Tension in lbs. per square in.	Compression in lbs. per square in.	Shearing strength in lbs. per sq. in.	Modulus of Elasticity in lbs. per square in.
ast iron	11,000 to 30,000	5,000 to 130,000		
ast iron, ave.	16,000	95,000	11,000	14,000,000 to 23,000,000
Vrought iron	40,000 to 70,000			26,000,000 to 31,000,000
Vrought iron, ave.	50,000	50,000	40,000	
oft steel, unhardened	60,000 to 100,000			
oft steel, ave.	80,000	70,000	55,000	30,000,000 to 36,000,000
oft steel, hardened	120,000			
ast steel, untempered	84,000 to 150,000			
ast steel, ave.	120,000			15,000,000 to 17,000,000
Vood, ash	17,000	9,300	1,400	
Vood, beech	11,500 to 22,000	7,700 to 9,300		
Vood, pine	11,000	6,000	650	1,400,000
Vood, oak	15,000	10,000	2,300	1,500,000
eather	4,200			25,000

TABLE 12.—SAFE LOADS IN TONS FOR SMALL STRUTS—(WROUGHT IRON).

Cross Section of Angles or Tees (Inches).	Length in feet. 3	4	6	8	10	12
2 x 2 x ¼	2.18	2.00	1.40	1.05	0.94	0.60
2 x 2 x ⅜	3.00	2.61	2.35	1.43	1.08	0.82
2½ x 2½ x ¼	2.71	2.43	1.93	1.43	1.14	0.85
2½ x 2½ x ⅜	4.00	3.55	2.81	2.10	1.65	1.24
3 x 3 x ¼	3.50	3.26	2.77	2.25	1.86	1.50
3 x 3 x ⅜	5.10	4.83	4.09	3.33	2.78	2.22
3 x 3 x ½	6.70	6.26	5.31	4.38	3.61	2.90
3½ x 3½ x ¼	4.21	4.00	3.33	2.66	2.21	1.7
3½ x 3½ x ⅜	6.15	5.85	5.03	4.22	3.56	2.90
3½ x 3½ x ½	8.10	7.54	6.74	5.80	4.90	4.04
4 x 4 x ⅜	7.24	7.00	6.32	5.52	4.81	4.10
4 x 4 x ½	9.28	9.00	8.06	7.22	6.26	5.37
4 x 4 x ⅝	11.70	11.20	10.60	8.90	7.75	6.43

TABLE 12a—DIAMETER, PITCH AND STRENGTH OF BOLTS.

Diameter in Inches	Diameter at Bottom of Thread	Threads to the Inch	Safe Load in Pounds
¼	0.185	20	323
5/16	0.240	18	542
⅜	0.294	16	737
7/16	0.344	14	1,118
½	0.400	13	1,515
9/16	0.454	12	1,940
⅝	0.507	11	2,424
¾	0.620	10	3,645
⅞	0.731	9	5,010
1	0.837	8	6,610
1⅛	0.940	7	8,350
1¼	1.065	7	10,100

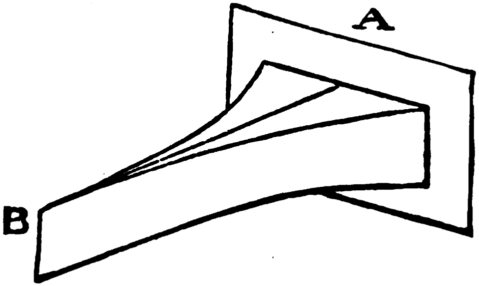


FIG. 60b.

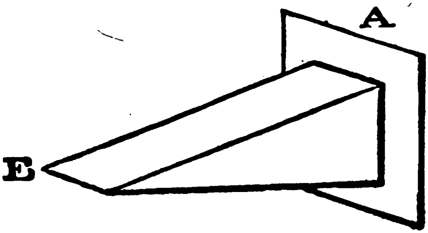


FIG. 60c.

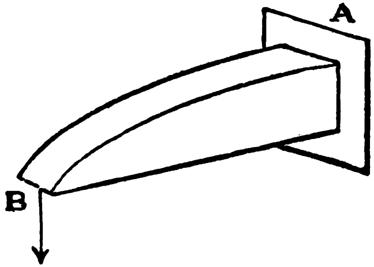


FIG. 60a.

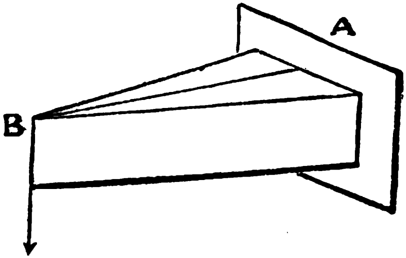


FIG. 60b.

Beams of Uniform Strength.—(RANKINE) are those in which the dimensions of the cross section are varied in such a manner, that its ultimate resistance bears at each point of the beam the same proportion to the moment of flexure. That resistance for figures of the same kind being proportional to the breadth, and to the square of the depth, can be varied, either by varying the breadth, the depth, or both. The law of variation depends upon the mode of variation of the moment of flexure of the beam from point to point, and this depends upon the distribution of the load, and of the supporting forces.

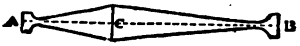


FIG. 61.

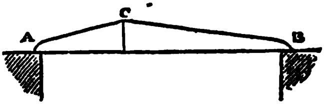


FIG. 61a.

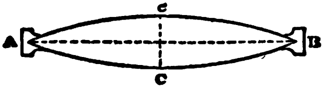


FIG. 62.

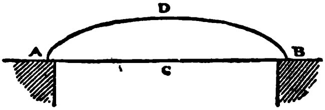


FIG. 62a.

When the depth of the beam is made uniform, and the breadth varied, the vertical longitudinal section is rectangular, and the plan is of a figure depending on the mode of variation of the breadth. When the breadth of the beam is made uniform, and the depth varied, the plan is rectangular, and the vertical longitudinal section is of a figure depending on the mode of variation of the depth. The following table gives examples of the results of those principles referring to illustrations;

Mode of Loading and Supporting.	bh^2, proportional to	Depth h constant; Figure ure of Plan.	Breadth b constant; Figure of Vertical Longitudinal Section.
I. (Figs. 60, 60a). Fixed at A, loaded at B,..	Distance from B.	Triangle, apex at B, fig. 60.	Parabola, vertex at B, fig. 60a.
II. (Figs, 60b, 60c). Fixed at A, uniformly loaded	Square of distance from B.	Pair of parabolas, vertices touching each other at B, fig. 60b.	Triangle, apex at B, fig. 60c.
III. Figs. 61, 61a). Supported at A and B, loaded at C, ...	Distance from adjacent point of support.	Pair of triangles, common base at C, apices at A and B, fig. 61.	Pair of parabolas, vertices at A and B, meeting at C, fig. 61a.
IV. (Figs. 62, 62a). Supported at A and B, uniformly loaded, ...	Product of distances from points of support.	Pair of parabolas, vertices at C, C, in middle of beam; common base A B, fig. 62.	Ellipse A D B, fig. 62a.

SHAFTING.

Power is usually delivered to various parts of a plant by means of shafting supported by hangers, and connected to the prime mover through a system of couplings, clutches and belts. Shafting may be either cold rolled steel, turned shafting, or wrought iron. Wrought iron shafting is little used at present, as the same power can be transmitted by much smaller and lighter steel shafting. In considering steel shafting it is greatly a matter of personal choice as to whether cold rolled steel or turned shafting be used. It will be found, however, that the cold rolled shafting has a slightly higher tensile strength than the turned shafting. This is due to the removal of the skin of the metal in turning. The one great advantage in turned shafting is that it will all be of the exact size wanted, and any flaws in the stock due to seams or the metal not being homogeneous, will be apparent, and the shaft, or the part of the shaft affected may be thrown out.

Strength of Shafting.—Table 13 will be found useful in calculating the strength of shafting. It has been calculated from the torsional moment of resistance of shafts, $M = \frac{\pi}{16} f\, d^3$, where $M =$ the torsional moment of resistance, $\pi = 3.1416$, $f =$ the stress per square inch to which the shaft is subjected. In the table two values are given for wrought iron, viz., 8000-lb. and 10,000-lb. d is the diameter of the shaft in inches. For cast iron, half the values given are to be taken, or 4,000-lb. and 5,000-lb. less per square inch. For the transverse or bending moment, the formula is $M = \frac{\pi}{32} f\, d^3$. In the case of a steam engine $M = P \times L$, where $P =$ total pressure on piston, and $L =$ length of crank in inches.

The formula $M = \frac{\pi}{16} f d^3$ when applied to an engine shaft becomes $P L = \frac{d^3}{5.1} f$; $d^3 = \frac{PL}{f} \times 5.1$, and $d = \sqrt[3]{\frac{PL}{f} \times 5.1}$, for twisting. For bending $d = \sqrt[3]{\frac{PL}{f} \times 10.2}$.

TABLE 18

Dia. of Shaft	f equals 8000lb.	f equals 10,000lb.	Dia. of Shaft.	f equals 8000lb.	f equals 10,000lb.	Dia. of Shaft.	f equals 8000lb.	f equals 10,000lb
in.			in.			in.		
1	1,570	1,962	4½	143,066	178,833	8	803,840	1,004,800
1¼	3,066	3,832	4¾	168,260	210,325	8½	964,176	1,205,220
1½	5,299	6,624	5	196,250	245,313	9	1,144,530	1,430,662
1¾	8,414	10,517	5¼	227,184	283,980	9½	1,346,079	1,682,599
2	12,560	15,700	5½	261,209	326,511	10	1,570,000	1,962,500
2¼	17,883	22,354	5¾	298,472	373,090	10½	1,817,471	2,271,839
2½	24,531	30,664	6	339,120	423,900	11	2,089,670	2,612,088
2¾	32,651	40,814	6¼	383,200	479,125	11½	2,387,774	2,989,717
3	42,390	52,988	6½	431,160	538,951	12	2,712,960	3,391,200
3¼	53,895	67,369	6¾	482,848	603,560	13	3,449,290	4,311,612
3½	67,314	84,143	7	538,810	673,138	14	4,308,080	5,385,100
3¾	82,793	103,491	7¼	598,293	747,866	15	5,298,750	6,623,428
4	100,480	125,600	7½	662,344	827,930	16	6,430,720	8,038,400
4¼	120,522	150,652	7¾	730,810	913,512	18	9,156,240	11,445,296

Shaft Bearings.—The distance apart of shaft bearings may be obtained by this rule: $L = 5\sqrt[3]{d^2}$ to $L = 4.8\sqrt[3]{d^2}$; the former value being used for shafting carrying its own weight only, and the latter when the ordinary proportion of pulleys, etc., are carried by the shaft.

$L =$ length in feet between supports.
$d =$ diameter of shaft in inches.

For a 4-in. shaft $L = 4.8 \sqrt[3]{16} = 12$ ft.

For diameters of shafts to resist torsional strains only, Molesworth gives the following formula: $d = \sqrt[3]{\frac{Pl}{K}}$, in which d = diameter in inches, P = twisting force in pounds applied at the end of a lever-arm whose length is l in inches, K = a coefficient whose values are: for cast iron 1,500, wrought iron 1,700, cast steel 3,200, gun-bronze 460 ,brass 425, copper 380, tin 220, lead 170. The value given for cast steel applies only to high-carbon steel.

Thurston gives, for head shafts well supported against springing, having bearings close to pulleys, or gears:

$$H.P. = \frac{d^3R}{125};\ d = \sqrt[3]{\frac{125\ H.P.}{R}},\ \text{for iron;}$$

$$H.P. = \frac{d^3R}{75};\ d = \sqrt[3]{\frac{75\ H.P.}{R}},\ \text{for cold rolled iron.}$$

For line shafting, hangers 8 feet apart:

$$H.P. = \frac{d^3R}{90};\ d = \sqrt[3]{\frac{90\ H.P.}{R}},\ \text{for iron;}$$

$$H.P. = \frac{d^3R}{55};\ d = \sqrt[3]{\frac{55\ H.P.}{R}},\ \text{for cold rolled iron.}$$

For transmission simply; no pulleys:

$$H.P. = \frac{d^3R}{62.5};\ d = \sqrt[3]{\frac{62.5\ H.P.}{R}},\ \text{for iron;}$$

$$H.P. = \frac{d^3 R}{35 R};\ d = \sqrt[3]{\frac{35\,H.P.}{R}}, \text{ for cold rolled iron}$$

in which H. P. = horsepower transmitted,

d = diameter of shaft in inches,

R = revolutions per minute.

Jones & Laughlin give the same formulæ as Dr. Thurston, except the following, for line shafting, hangers 8 ft. apart:

$$H.P. = \frac{d^3 R}{50};\ d = \sqrt[3]{\frac{50\,H.P.}{R}}, \text{ for cold rolled iron.}$$

For simple transmission, and short contours:

$$H.P. = \frac{d^3 R}{50};\ d = \sqrt[3]{\frac{50\,H.P.}{R}}, \text{ for turned iron;}$$

$$H.P. = \frac{d^3 R}{30};\ d = \sqrt[3]{\frac{30\,H.P.}{R}}, \text{ for cold rolled iron.}$$

As the strain upon a shaft caused by a load is proportional to the products of the parts of the shaft multiplied into each other, therefore if the load be applied near one end of the span or bay instead of at the center, multiply the fourth power of the diameter of the shaft required to carry the load at the center of the bay by the product of the two parts of the shaft when the load is near one end, and divide this product by the product of the two parts of the shaft, when the load is carried at the center. The fourth root of this quotient will be the diameter required. The shaft in a line which carries a receiving pulley, or which carries a transmitting pulley to drive another line, should always be considered a head shaft, and should be of the size given by the rules for shafts carrying main pulleys, or gears.

For continuous shafting so proportioned as to deflect not more than .01 inch per foot of length, allowance being made for the weakening effect of key-seats, the following formulæ is given:

$$d = \sqrt[3]{\frac{50\ \text{H.P.}}{R}};\ L = \sqrt[3]{720d^2},\ \text{for bare shafts;}$$

$$d = \sqrt[3]{\frac{70\ \text{H.P.}}{R}};\ L = \sqrt[3]{140d^2}\ \text{for shafts carrying pul-}$$

leys, etc., in which $d =$ diameter in inches,
$L =$ length in feet,
$R =$ revolutions per minute.

Table 14 (by J. B. Francis) gives the greatest admissible distances between the bearings of continuous shafting, subject to no transverse strain, except from their own weight, as would be the case if the power given off from the shaft was equal on all sides, and at an equal distance from the hanger bearings.

TABLE 14.—DISTANCE BETWEEN BEARINGS.

Diam. of Shaft in Inches	Wrought Iron Shafts	Steel Shafts
2	15.46 feet	15.89 feet
3	17.70 feet	18.19 feet
4	19.48 feet	20.02 feet
5	20.99 feet	21.57 feet
6	22.30 feet	22.92 feet
7	23.48 feet	24.13 feet
8	24.55 feet	25.23 feet
9	25.53 feet	26.24 feet

Hollow Shafts.—Let $d =$ the diameter of a solid shaft, and d_1 d_2 the external and internal diameters of a hollow shaft of the same material; then the two shafts

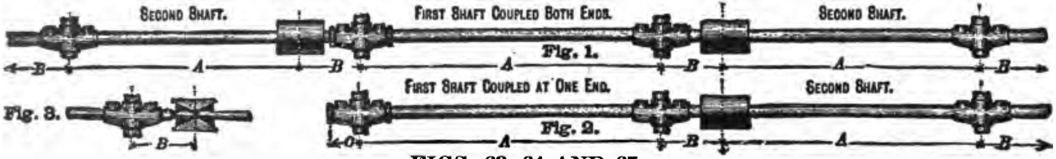
TABLE 15 FOR LAYING OUT SHAFTING.

	Nominal Size of 2d Shaft	1½"	1¾"	2"	2¼"	2½"	2¾"	3"	3¼"	3½"	4"	4½"	5"	5½"	6"	6½"	7"	7½"	8"	Length of Bearing, or Box, ins.	Double Cone-vise Coupling	
Length of Collared End for Fast Coll., ins.	Nominal Size of 1st Shaft, ins.	Distance from Center of Bearing to End of Shaft for Coupling. See B, Figs. 63, 64 and 65.																			Length inches	Diameter inches
3½	1½	8½																		6	5½	3¾
4	1¾	9	9½																	7	6½	4⅝
4⅜	2	9½	10½	11																8	7⅝	5⅜
5	2¼		11	11½	12															9	8¼	5⅞
5⅝	2½		11½	12	12½	13														10	9½	6 9/16
6¼	2¾		12	12½	13	13½	14													11	10	7
6¾	3			13	13½	14	14½	15												12	11	7¾
7¼	3¼			13½	14	14½	15	15½	15½											13	12	8¼
7⅞	3½			14	14½	15	15½	16	16	16½										14	13	9 1/16
8 15/16	4			15	15½	16	16½	17	17	17½	18									16	14¼	10
10	4½				16½	17	17½	18	18	18½	19½	20								18	16	11⅜
11⅛	5					18½	18½	19	19½	20	20½	21½	22							20	18	12 9/16
12 3/16	5½						20	20½	21	21	22	22½	23½	24						22	19	13½
13¼	6						21½	21½	22	22½	23	24	24½	25	26					24	21	14½
14⅜	6½							23	23½	24	24½	25	26	26½	27½	28½				26	25	17⅜
15½	7								25	25	26	26½	27½	28	29	30	30½			28	25	17⅜
16½	7½									26½	27	28	28½	29½	30	31	31½	32		30	28⅜	19 5/16
17¾	8										28½	29½	30	31	31½	32½	32	34	34½	23	28⅜	19 5/16

Use of Table.—Look for size of first shaft in left-hand column, under the head of Size of first shaft, and in the top line of table, marked Size of second shaft, find the size of the shaft to be coupled to it. The intersection gives the length B; this added to the length A, or distance from center to center of bearing, and in cases similar to Fig. 64, to the length C, gives the length of the first shaft, thus: as in Fig. 63, B+A+B=length; Fig. 64, C+A+B=length.

Make bearings at equal distances from each other, when practicable, and always put two bearings on the first, which is the collared shaft. See Figs. 63 and 64.

In coupling shafts of different sizes, either reduce the end of the large shaft in diameter, and use a small coupling, or use a coupling to suit the larger shaft, with one cone bored for the smaller nominal shaft.



FIGS. 63, 64 AND 65.

will be of equal torsional strength when $d^3 = \dfrac{d_1^4 - d_2^4}{d_1}$.

A 10-inch hollow shaft with internal diameter of 4 inches will weigh 16 per cent less than a solid 10-inch shaft, but its strength will be only 2.56 per cent less. If the hole were increased to 5 inches, the weight would be 25 per cent less than that of the solid shaft, and the strength 6.25 per cent less. The following table is used by Wm. Sellers & Co. to facilitate the laying out of shafting. Figs. 63, 64 and 65 show the position of the hangers and position of coupling, for use either in case of extension in both directions from a central head-shaft, or extension in one direction from that head-shaft. The speed of shafting depends upon the kind of machinery it is used to transmit power to. The usual speed is as follows:

	R. P. M.
For Machine shops	120 to 180
" Wood-working	250 to 300
" Cotton and woolen mills	300 to 400

Deflection of Shafting—(Pencoyd Iron Works).—As the deflection of steel and iron is practically the same under similar conditions of dimensions and loads, and as shafting is usually determined by its transverse stiffness, rather than its ultimate strength, practically the same dimensions should be used for steel, as for iron. For continuous line-shafting it is considered good practice to limit the deflection to a maximum of .01 of an inch per foot of length. The weight of bare shafting in pounds is: $W = 2.6d^2L$, or as when fully loaded with pulleys, which is customary in practice, and allowing 40 lbs. per inch of width for the vertical pull of the belts, experience shows the load in pounds to be about $W = 13d^2L$. Taking the modulus of transverse elasticity at 26,000,000 lbs., we derive from authoritative formulæ the following:

$$L = \sqrt[3]{873d^2},\ d = \sqrt{\frac{L^3}{873}}$$ for bare shafting;

$$L = \sqrt[3]{175d^2},\ d = \sqrt{\frac{L^3}{175}}$$ for shafting carrying pulleys, etc., in which L = maximum distance in feet between bearings for continuous shafting subjected to bending stress alone, and d = diameter in inches.

Torsion in a Shaft.—The torsional stress in a shaft is inversely proportional to the velocity of rotation, while the bending stress will not be reduced to the same ratio. The amount of torsion in any shaft should be no more than just enough to permit a deflection of not more than one degree in thirty diameters of the shaft.

For instance, a shaft 2 inches in diameter by 5 feet in length should not twist more than one degree when under full load, with the power applied at one end and given off at the other end of the shaft.

This means that the 2-in. shaft would require to be 1,800 ft. in length before the allowable torsion or twist would allow it to make one complete revolution.

The load upon any shaft is termed the "twisting moment," and it is found by adding together the products of the radii of all the pulleys after each has been multiplied by the belt pull upon each pulley.

All of the pulleys that take power from the shaft are to be thus treated, while the pulley which delivers power to the shaft is to be treated in a similar manner, but by itself, and the product of that belt pull, and the radius of its pulley will be found equal to the sum of all the other belt-radiï products. Thus the greatest twisting moment will be found between the driving pulley and the power pulley next adjoining that pulley; but the bending or twisting strain may be divided at the driving pulley by a portion of the stress being delivered to pulleys on either side of the drive pulley.

The twisting moment, or torsional stress, is purely a shearing stress, as illustrated in a flange coupling, where the strain tends to shear off the coupling bolts. This stress is resisted by the "resisting moment," which should equal the sum of all the moments of shearing stresses about the axis of the shaft. Let $S =$ working torsional strength of the shaft material, $d =$ diameter of shaft, $n =$ number of revolutions per minute; then the horsepower that any shaft is capable of transmitting is $\text{H.P.} = \dfrac{S\,d^3 n}{321000}$. This formula is based upon the assumption that where $M =$ twisting moment, $0.1963\ S \times d^3 = M$.

The bending moment of a shaft is figured exactly as that of a beam, and the weight of the shaft itself constitutes a uniform load, while the weights of the pulleys, and pull of the belts constitute position loads, and should be figured accordingly when calculating the diameter of a shaft required for a given location. The stretch of a shaft depends upon the modulus of elasticity of the material from which it is made. Table 16 gives the modulus (or coefficient) of elasticity of four materials with which the millwright is brought most intimately in contact:

TABLE 16

Material.	Modulus.
Steel	30,000,000 lbs. per sq. in.
Wrought iron	27,500,000 " " " "
Cast iron	15,000,000 " " " "
Timber	1,800,000 " " " "

Problem: What will be the stretch of a steel rod 20 ft. long, and $\frac{7}{8}$-in. in diameter, loaded with 5,000 lbs. pull?

Solution: $D = \frac{PL}{SE}$, in which D = stretch of rod;

P = pull, L = length, S = sectional area of rod, E = modulus of elasticity for steel. The length L is to be expressed in inches as follows: $L = 20 \times 12 = 240$ inches, also the sectional area which is 0.6 inches then

$$\frac{5{,}000 \times 240}{0.6 \times 30{,}000{,}000} = 0.0667$$ inches, which is the stretch of the rod under the given conditions. The same rule applies to the torsion of shafts, and the moduli of elasticity for shear are about two-fifths of the standard moduli, as for instance, for steel, 11,000,000 lbs.; for wrought iron, 10,000,000 lbs.; and for cast iron, 6,000,000 lbs. Let E^1 represent the modulus of elasticity for shear, the angle a, through which a shaft 2 in. in diameter and 20 feet long will be twisted, is found as follows: using the same formula in which the symbols have a similar meaning, S in this case being 10031, and n 200, we have $H.P. = \frac{S\,d^3 n}{321000} = 50$ horsepower, and

$$a = \frac{584\ TL}{E^1\ d^3} = \frac{36{,}800{,}000\ H.P.L.}{E^1\ d^4\ n} =$$

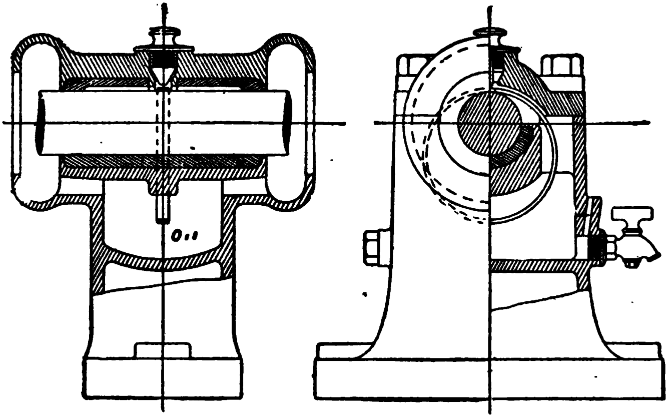
$$\frac{36{,}800{,}000 \times 50 \times 240}{11{,}000{,}000 \times 16 \times 200} = 12\tfrac{1}{2}\ \text{degrees.}$$

The number of degrees that this shaft could be twisted with safety is 4. Therefore it is evident from the above calculation that 50 H. P. is an overload, and that the strain should be reduced, or a larger shaft be put in. Taking 4 as the permissible number of degrees the shaft may twist, the safe load is found as follows: the shaft is 240 ÷ 2 = 120 diameters in length, and 120 ÷ 4 = 30 H. P. as the safe load.

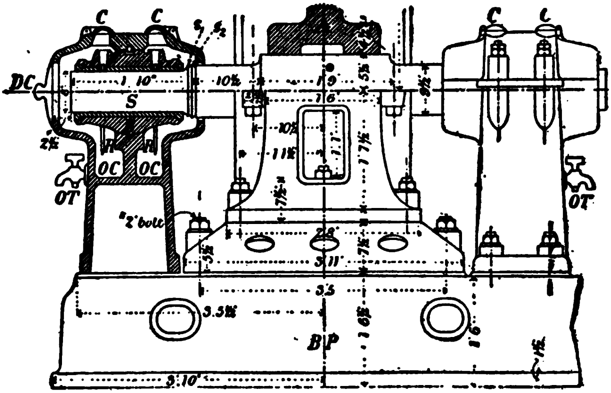
It is evident from the above that the safe length of a shaft depends upon its diameter, the number of pulleys it is to carry, and the manner in which the shaft bearings are arranged. The shafting is delivered by the manufacturer in various lengths up to 40 feet, and it is not good policy to have the several lengths which go to make up a line shaft too long for convenience in handling. Lengths of shafts which, when loaded with their pulleys and gears, will bend under their own weight when hung up from two points, are too long to be handled with safety.

In the designing of a line of shafting, this matter should receive attention, and a sufficient number of couplings be placed to keep the lengths to such dimensions that they may be conveniently handled when the pulleys and gears have been put in place.

Shafts may be solid or hollow. Hollow shafts are stronger than solid shafts for the same weight of material.



SINGLE RING SELF-OILING BEARING.



TWO RING SELF-OILING BEARING.
FOR A LARGE ALTERNATE CURRENT DYNAMO.

FIG. 66.

Box Couplings. The coupling illustrated in Fig. 67 consists of a solid box made of cast-iron, bored out to fit the shafts, whose ends are made to butt together inside the box. The box may be secured to the shafts by means of a sunk key which extends the whole length of the box. A better arrangement is to use two keys, both driven from the same end of the box, as shown in Fig. 67. With two keys it is not so important that the keyways in the shafts have exactly the same depth. Two keys may be more tightly driven in, and are easier to drive back than a single key of the same total length. There must, however, be a clearance space between the head of the forward key, and the point of the hind one, so as to ensure that the latter is driven in with the same degree of tightness as the former. In driving the keys back also, they can be started separately, and therefore more easily, when this clearance space is allowed. Fig. 67 shows the ends of the shafts enlarged where the keyways are cut, so that the latter do not weaken the shafts, the amount of the enlargement being such that the bottom of the keyway touches the outside of the shaft.

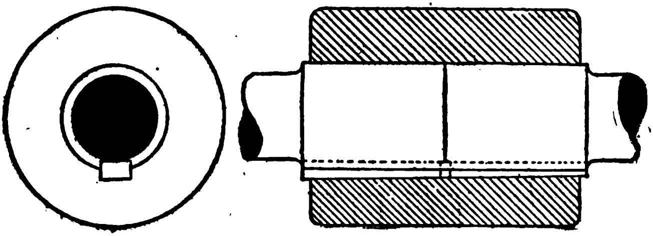


FIG. 67.
BOX SHAFT COUPLING.

Split Couplings. This form of coupling, which is shown in Fig. 67, is very easily put on or taken off, it has no projecting parts, the bolts being completely covered, and, like the solid couplings, it may be used as a driving pulley, or a driving pulley may be placed on it. In making this coupling the faces for the joint between the two halves of the box are first planed. The bolt-holes are then drilled and the two halves bolted together with pieces of paper between them; then the muff is bored out to the exact size of the shaft. When the paper is removed and the box put on the shaft and bolted up, the box grips the shaft firmly. The key has no taper and should fit on the sides only.

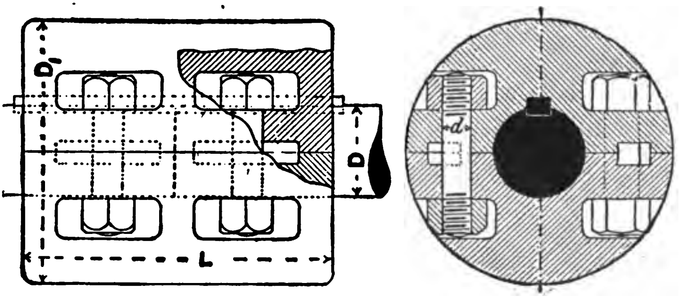


FIG. 68.

SPLIT SHAFT COUPLING

DIMENSIONS OF SPLIT COUPLINGS.

Diam. of shaft, D	1½	1¾	2	2¼	2½	2¾	3	3½	4
Diam. of Box, D1	4½	5¼	5¾	6	6½	7¼	7⅞	9¾	10½
Length of Box, L	6	7	8	9	10	11	12	14	16
Diam. of Bolts, d	½	⅝	⅝	⅝	¾	⅞	⅞	⅞	⅞
Number of Bolts	4	4	4	4	4	4	4	6	6

Cast-Iron Flange Coupling. In this form of coupling, which is shown in Fig. 69, there are cast-iron pieces or bosses, provided with flanges which are keyed to the ends of the shafts to be connected. These flanges are fastened together by means of bolts and nuts as shown.

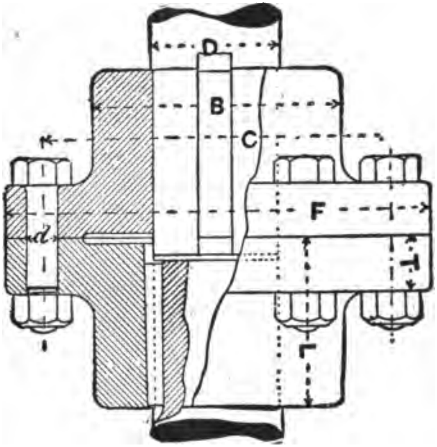


FIG. 69.
CAST-IRON FLANGE COUPLING.

Sometimes the shaft is enlarged where it enters the coupling, so as to allow for the weakening effect of the keyway, but more frequently it is parallel throughout, or very slightly reduced, as shown in Fig. 69, so as to form a shoulder, which prevents the shaft going farther into the coupling.

DIMENSIONS OF CAST-IRON FLANGE COUPLINGS.

Diam. of Shaft D.	Diam. of Flange F.	Thickness of Flange T.	Diam. of Boss B.	Depth at Boss L.	Number of Bolts	Diam. of Bolts d.	Diam. of Bolt Circle C.
1	5 5-8	3-4	2 5-8	2 1-16	3	1-2	4 1-4
1 1-4	6 13-16	13-16	3 1-16	2 5-16	3	5-8	5 1-16
1 1-2	7 1-4	7-8	3 1-2	2 5-8	3	5-8	5 1-2
1 3-4	7 11-16	1	3 15-16	2 7-8	4	5-8	5 15-16
2	8 7-8	1 1-16	4 3-8	3 3-16	4	3-4	6 3-4
2 1-4	9 1-16	1 1-8	4 3-4	3 7-16	4	3-4	7 1-8
2 3-4	11	1 5-16	5 3-4	4 1-16	4	7-8	8 9-16
3	12 3-8	1 7-16	6 1-4	4 5-16	4	1	9 1-2
3 1-4	12 5-8	1 1-2	6 5-8	4 5-8	4	1	9 13-16
3 1-2	13 1-8	1 5-8	7 1-8	4 7-8	4	1	10 5-16
3 3-4	13 9-16	1 11-16	7 9-16	5 3-16	4	1	10 3-4
4	14	1 3-4	8	5 7-16	6	1	11 1-4
4 1-4	14 7-16	1 7-8	8 7-16	5 3-4	6	1	11 5-8
4 1-2	15 5-8	2	8 7-8	6	6	1 1-8	12 1-2
4 3-4	16 1-8	2 1-16	9 3-8	6 5-16	6	1 1-8	13
5	17 5-16	2 1-8	9 13-16	6 5-8	6	1 1-4	13 13-16
5 1-4	17 3-4	2 1-4	10 1-4	6 7-8	6	1 1-4	14 1-4
5 1-2	18 3-16	2 5-16	10 3-4	7 1-4	6	1 1-4	14 11-16
5 3-4	19 1-2	2 7-16	11 1-4	7 7-16	6	1 3-8	15 5-8

MATERIALS USED IN MACHINE CONSTRUCTION.

Cast Iron. The essential constituents of cast iron are iron and carbon, the latter forming from 2 to 5 per cent. of the total weight. Cast iron, however, usually contains varying small amounts of silicon, sulphur, phosphorus, and manganese.

In cast iron the carbon may exist partly in the free state and partly in chemical combination with the iron.

Chilled Castings. When grey cast iron is melted a portion of the free carbon combines chemically with the iron, this, however, separates out again if the iron is allowed to cool slowly, but if it is suddenly cooled a greater amount of the carbon remains in chemical combination, and a whiter and harder iron is produced. Advantage is taken of this in making chilled castings. In this process the whole or part of the mould is lined with cast iron, which, being a comparatively good conductor of heat, chills a portion of the melted metal next to it, changing it into a hard white iron to a depth varying from ⅛ to ½ an inch. To protect the cast iron lining of the mould from the molten metal it is painted with loam.

Malleable Cast Iron. This is prepared by imbedding a casting in powdered red hematite, an oxide of iron, and keeping it at a bright red heat for a length of time varying from several hours to several days according to the size of the casting. By this process a portion of the carbon in the casting is removed, and the strength and toughness of the latter become more like the strength and toughness of wrought iron.

Wrought Iron. This is nearly pure iron, and is made from cast iron by the puddling process, which consists chiefly of raising the cast iron to a high temperature in a reverberatory furnace in the presence of air, which unites with the carbon and passes off as gas. In other words the carbon is burned out. The iron is removed from the puddling furnace in soft spongy masses called blooms, which are subjected to a process of squeezing or hammering called shingling. These shingled blooms still contain enough heat to enable them to be rolled into rough puddled bars. These puddled bars are of very inferior quality, having less than half the strength of good wrought iron. The puddled bars are cut into pieces which are piled together, reheated, and again rolled into bars, which are called merchant bars. This process of piling, reheating, and re-rolling may be repeated several times, depending on the quality of iron required. Up to a certain point the quality of the iron is improved by reheating and rolling or hammering, but beyond that a repetition of the process diminishes the strength of the iron.

TABLE 17

HORSE-POWER OF SHAFTING.

AS MAIN SHAFTING, CARRYING DRIVING PULLEYS OR GEARS.

Diameter of Shaft in Inches.	Number of Revolutions per Minute.										
	60	80	100	125	150	175	200	225	250	275	300
1¾	2·6	3·4	4·3	5·4	6·4	7·5	8·6	9·7	10·7	11·8	12·9
2	3·8	5·1	6·4	8	9·6	11·2	12·8	14·4	16	17·6	19·2
2¼	5·4	7·3	8·1	10	12	14	16	18	20	22	24
2½	7·5	10	12·5	15	18	22	25	28	31	34	37
2¾	10	13	16	20	24	28	32	36	40	44	48
3	13	17	20	25	30	35	40	45	50	55	60
3¼	16	22	27	34	40	47	54	61	67	74	81
3½	20	27	34	42	51	59	68	76	85	93	102
3¾	25	33	42	52	63	73	84	94	105	115	126
4	30	41	51	64	76	89	102	115	127	140	153
4½	43	58	72	90	108	126	144	162	180	198	216
5	60	80	100	125	150	175	200	225	250	275	300
5½	80	106	133	166	199	233	266	299	333	366.	400

AS SECOND MOTION OR LINE SHAFTING.

Bearings Eight Feet Apart.

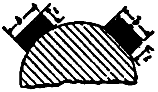
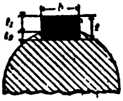
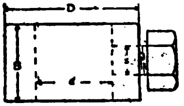
Diameter of Shaft in Inches.	Number of Revolutions per Minute.										
	100	125	150	175	200	225	250	275	300	325	350
1¾	6	7·4	8·9	10·4	11·9	13·4	14·9	16·4	17·9	19·4	20·9
1⅞	7·3	9·1	10·9	12·7	14·5	16·3	18·2	20	21·8	23·6	25·4
2	8·9	11·1	13·3	15·5	17·7	20	22·2	24·4	26·6	28·8	31
2⅛	10·6	13·2	15·9	18·5	21·2	23·8	26·5	29·1	31·8	34·4	37
2¼	12·6	15·8	19	22	25	28	31	35	38	41	44
2⅜	15	18	22	26	29	33	37	41	44	48	52
2½	17	21	26	30	34	39	43	47	52	56	60
2¾	23	29	34	40	46	52	58	64	69	75	81
3	30	37	45	52	60	67	75	82	90	97	105
3¼	38	47	57	66	76	85	95	104	114	123	133
3½	47	59	71	83	95	107	119	131	143	155	167
3¾	58	73	88	102	117	132	146	162	176	190	205
4	71	89	107	125	142	160	178	196	213	231	249

FOR SIMPLY TRANSMITTING POWER.

Diameter of Shaft in Inches.	Number of Revolutions per Minute.										
	100	125	150	175	200	233	267	300	333	367	400
1½	6·7	8·4	10·1	11·8	13·[illegible]	15·7	17·9	20·3	22·5	24·8	27
1⅝	8·6	10·7	12·8	15	17·1	20	22·8	25·8	28·6	31·5	34·3
1¾	10·7	13·4	16	18·7	21·5	25	28	32	36	39	43
1⅞	13·2	16·5	19·7	23	25·4	31	35	39	44	48	52
2	16	20	24	28	32	37	42	48	53	58	64
2⅛	19	24	29	33	38	44	51	57	63	70	76
2¼	22	28	34	39	45	52	60	68	75	83	90
2⅜	27	33	40	47	53	62	70	79	88	96	105
2½	31	39	47	54	62	73	83	93	104	114	125
2¾	41	52	62	73	83	97	111	125	139	153	167
3	54	67	81	94	108	126	144	162	180	198	216
3¼	68	86	103	120	137	160	182	205	228	250	273
3½	85	107	128	150	171	200	228	257	285	313	342

TABLE 18

DIMENSIONS OF KEYS AND THRUST COLLARS.

Diameter of Shaft in Inches.	Keys.					Cast-Iron Collars.				Diameter of Shaft in Inches.
										
d	*b*	*t*	*t*	*t*$_1$	*t*$_2$	D	B	*s*	No. of Screws.	*d*
1	3/8	5/32	1/4	5/32	3/32	1 3/4	1	3/8	1	1
1 1/4	7/16	3/16	9/32	7/16	3/32	2 1/8	1	„	„	1 1/4
1 1/2	1/2	7/32	5/16	7/32	3/32	2 1/2	1 1/4	1/2	„	1 1/2
1 3/4	9/16	1/4	11/32	7/32	1/8	2 7/8	1 7/16	„	„	1 3/4
2	5/8	9/32	3/8	1/4	1/8	3 1/8	1 5/8	5/8	„	2
2 1/4	11/16	5/16	13/32	1/4	5/32	3 1/2	1 7/8	„	„	2 1/4
2 1/2	3/4	5/16	7/16	9/32	5/32	3 3/4	2	„	„	2 1/2
2 3/4	13/16	11/32	7/16	9/32	5/32	4 1/8	2 3/16	„	„	2 3/4
3	7/8	3/8	1/2	5/16	3/16	4 3/8	2 3/8	3/4	„	3
3 1/4	15/16	3/8	1/2	5/16	3/16	4 3/4	2 1/2	„	„	3 1/4
3 1/2	1	13/32	9/16	3/8	3/16	5 1/8	2 3/4	„	„	3 1/2
3 3/4	1 1/16	13/32	9/16	3/8	3/16	5 1/2	2 7/8	„	„	3 3/4
4	1 1/8	7/16	5/8	13/32	7/32	5 3/4	3	„	2	4
4 1/4	1 3/16	1/2	5/8	13/32	7/32	6 1/8	3 1/4	„	„	4 1/4
4 1/2	1 1/4	1/2	11/16	7/16	1/4	6 1/2	3 1/2	„	„	4 1/2
4 3/4	1 5/16	9/16	11/16	7/16	1/4	6 3/4	3 5/8	„	„	4 3/4
5	1 3/8	9/16	3/4	1/2	1/4	7 1/8	3 3/4	„	„	5
5 1/4	1 7/16	19/32	3/4	1/2	1/4	7 1/2	4	„	„	5 1/4
5 1/2	1 1/2	5/8	13/16	17/32	9/32	7 3/4	4 1/8	„	„	5 1/2
5 3/4	1 9/16	5/8	13/16	17/32	9/32	8 1/8	4 3/8	„	„	5 3/4
6	1 5/8	21/32	7/8	19/32	9/32	8 1/2	4 1/2	„	„	6
6 1/4	1 11/16	11/16	7/8	19/32	9/32	8 3/4	4 5/8	7/8	„	6 1/4
6 1/2	1 3/4	11/16	15/16	5/8	5/16	9	4 3/4	„	„	6 1/2
6 3/4	1 13/16	3/4	15/16	5/8	5/16	9 3/8	5	„	„	6 3/4
7	1 7/8	3/4	1	11/16	5/16	9 3/4	5 1/4	„	„	7
7 1/4	1 15/16	13/16	1	11/16	5/16	10	5 3/8	„	„	7 1/4
7 1/2	2.	13/16	1 1/16	3/4	5/16	10 3/8	5 1/2	„	„	7 1/2
7 3/4	2 1/16	7/8	1 1/16	3/4	5/16	10 3/4	5 3/4	„	„	7 3/4
8	2 1/8	7/8	1 1/8	3/4	3/8	11 1/4	6	„	„	8

Erection of Shafting.—When it is necessary to erect piers for supporting a line of shafting, whether the piers be of concrete, stone, or brick, one point should never be lost sight of, viz: that in addition to the dead weight of the shafting and pulleys, there will also be the strains caused by the pull of the belts, some in one direction and some in another, and these strains do not come on all the piers alike, some being called upon to withstand more than others. Therefore, the proper thing to do is to design each pier with a view to the strain that is likely to come upon it. For instance, let Fig. 71 represent a line of shafting located 6 ft. above the floor line, and it is required to ascertain the dimensions and shape of each of the several piers, marked I, II, III, V and VI.

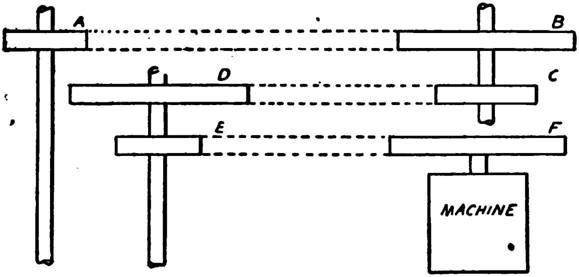


FIG. 70
REDUCING PULLEY SPEED IN SEVERAL STEPS.

For present purposes the length of the shaft is assumed to be 34 ft., and the diameter is $3\frac{15}{16}$ inches. The weight of the shaft and pulleys being known, it remains to ascertain the strains that will result from the pull of the belts.

It is customary to limit the force of this pull to 400 lbs. per inch width of belt, and by reference to Fig. 71, it will be seen that there is a pulley D in close proximity to pier VI. The width of the belt running over this pulley is 14 inches. There will therefore be 400 × 14 = 5600 lbs. exerted in a horizontal direction at a height of 6 feet from the floor line, illustrated at a, Fig 72. Should it be necessary to go to a depth of 4 feet in order to secure a suitable bearing soil for the foot of the pier, the force of 5,600 lbs. would be exerted through a lever $6 + 4 = 10$ feet in length, and as the force in pounds applied at the end of a lever multiplied by the length of the lever in feet reduces the leverage to foot pounds of statical moment, the moment of the force of 5600 lbs. will be $5600 \times 10 = 56000$ lbs.

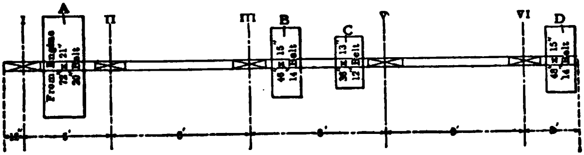


FIG. 71.
SHAFT SUPPORTING PIERS.

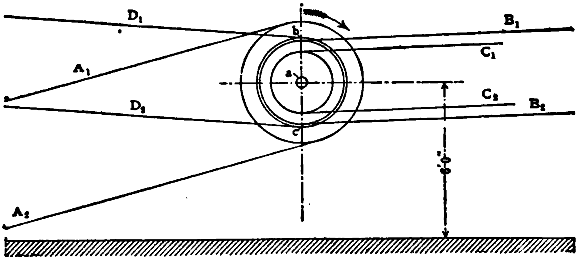


FIG. 72.
BELT PULL ON SHAFT SUPPORTING PIERS.

To resist this force, two factors are to be considered; first, weight of the pier, and second length of foundation at bottom. This is illustrated in Fig. 73 to which reference is now made. If the foundation there shown was only one foot in length at its base and weighed 56000 lbs., it would just balance the pull of the belt, and if it were made two feet long, as from d to f for instance, its weight could be reduced one-half, or 56,000 ÷ 2 = 28,000 lbs. and it would still balance the belt pull, provided the weight could all be applied at d, which cannot be done because the weight (28000 lbs.) is distributed over the whole area of the base extending to point f, which is in fact the fulcrum.

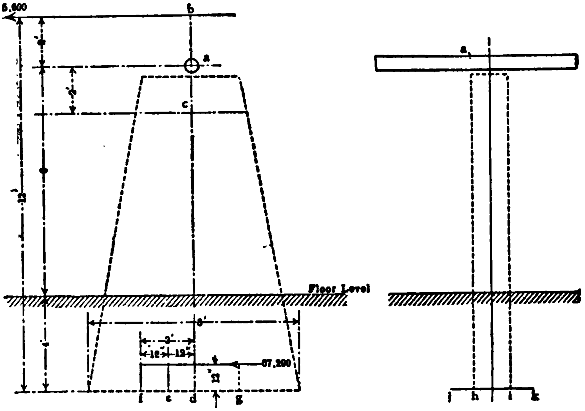


FIG. 73.
CALCULATING WEIGHT NECESSARY IN A PIER.

For this reason the actual holding down weight or power of the pier in its present form is only one-half what it appears to be, and it becomes necessary to increase its length to point g, Fig. 73, thus making it 4 feet in length at right angles to the shaft and an effective weight of 28000 lbs.

The width of the pier parallel with the shaft depends upon the nature of the soil or footing, and the total weight required. The figures given above merely show the net weight required to balance the pull of the belt on pier VI (Fig. 71), assuming this pull to be exerted at a, Fig. 73, but as pulley D is 4 feet in diameter, there should be one-half of this diameter added to the length of the lever, making it 12 instead of 10 feet, in length, and with a moment equal to 5600 × 12 = 67200 lbs., and the actual holding down weight required from d to g will be 67,200 ÷ 2 = 33,600 lbs. From this may be deducted the combined weights of:

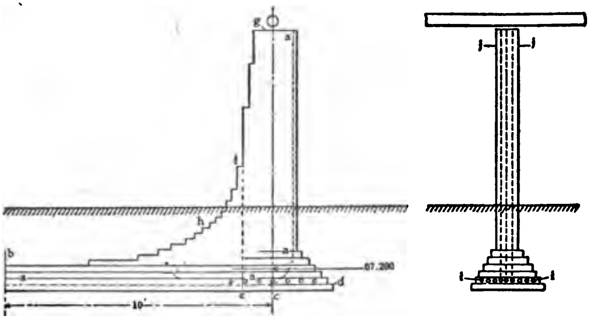


FIG. 74.
ECONOMY OF REINFORCED CONCRETE.

7 feet of shaft, 3$\frac{15}{16}$ inches in diam.	288 lbs.
The pillow block	100 lbs.
Four bolts	25 lbs.
1 Pulley 48 x 15 inches	550 lbs.
30 feet of 14 inch 6 ply belt	140 lbs.
Total	1103 lbs.

The weight of masonry necessary will then be 33600 — 1103 = 32497 lbs. The weight of the material to be used will range from 140 lbs. per cu. ft. for brick work, to 160 lbs. per cu. ft. for concrete. If the pier be built of brick, there will be required 32497 ÷ - 140 = 232 cu. feet of masonry. As the pier is to be

10 feet high, it is obvious that the cross-sectional area must be 232 ÷ 10 = 23.2, and if the pier is to be made 4 feet long, its width must be about 5 feet 10 inches, or wider than it is long.

It is obvious that the above noted dimensions are not very suitable for the pier in question. Therefore, the length may be increased to 8 feet, which will double the turning resistance, while at the same time the holding down weight may again be reduced one-half or 33600 ÷ 2 = 16800 lbs. Deducting the weight of shaft, pulley, etc., from this, will leave 16,800 — 1,103 = 15,697 lbs. of masonry required, equal to 15697 ÷ 140 = 112 cu. feet.

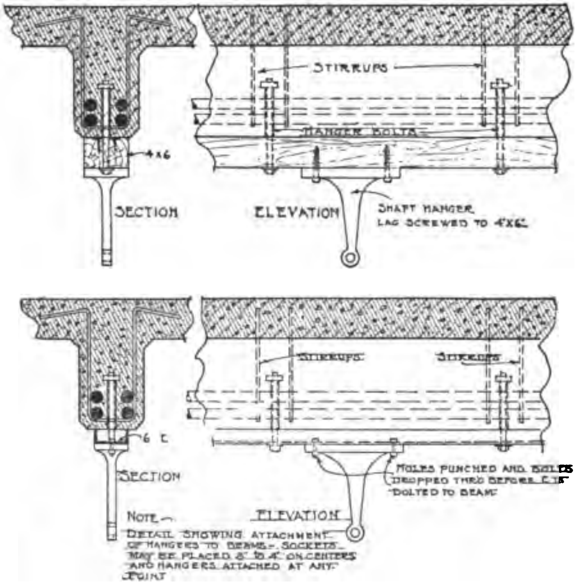


FIG. 75.

If the pier be made 8 feet in length at the foot, and tapering to 4 feet in length at the top, the average length would be 6 feet, giving a surface area of $10 \times 6 = 60$ square feet, and as 112 cu. feet of masonry are required, the width should be $112 \div 60 = 1\ 5/6$ feet or say 2 feet.

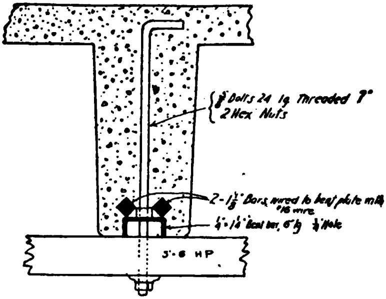

7/8" Bolts 24 lg Threaded 7"
2 Hex Nuts
2-1 1/8" Bars wired to bearing plate with #16 wire
1/4" × 1 1/4" Bearing bar, 6" lg 1/2" Hole
3'-6 HP

FIG. 76.

The foot of the pier has an area of $8 \times 2 = 16$ sq. feet, and as the load is about 8 tons, there will be one-half ton per sq. foot which is a safe load without any extension of footing.

Both side and end elevation are outlined in Fig.

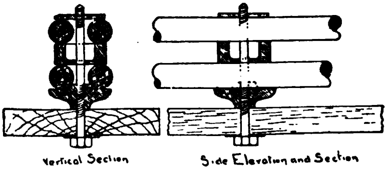


FIG. 77.

73. Another type of pier may be constructed as shown in Fig. 74, in which the quantity of masonry is reduced from 112 to 6 cubic feet. In this form of construction, the bearing for the 67,200-pound thrust moment 12 inches above the bottom of the foundation is

carried 10 feet away toward b, and instead of having been able to divide the thrust moment by 4, as in Fig. 73, we are now able to divide it directly by 10, the footing at b, and intermediate between b and c being sufficient to absorb the pressure.

The division by 10 allows the belt pull to be balanced by a masonry load of 6720 pounds, and deducting the shaft-load of 1103 pounds there remains only 5617 pounds weight of masonry necessary, or a trifle more than 40 cubic feet for the pillar g, e.

The cross section will now be 40 ÷ 10 = 4 sq. feet, which corresponds to its being 4 feet long and 12 inches thick. This takes no account of the brace that extends to b, more than half of which in weight is accountable as being attached to g, e, thereby increasing the resistance of the pier. There is also the holding power of the soil which may be taken into account if necessary.

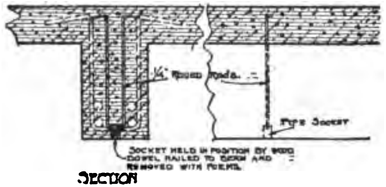

¼" Round Rods.
Pipe Socket
Socket held in position by wood dowel nailed to beam and removed with forms.
Section

FIG. 78.

In order to give the required strength to withstand the transverse or breaking strain due to the pull of the belt, the pier is reinforced with steel bars placed as shown at a, a, a, a, Fig. 74. The strength of steel necessary at any point between b and c may be found by considering its distance from c as the length of a lever which is to be divided into the moment—67,200. Thus at a point 6 feet from c, toward b, the load will be

67,200 ÷ 6 = 11,200 pounds, which may be considered as transverse strain tending to break the brace foundation at a point 6 feet from c.

ATTACHING SHAFTING AND MACHINERY IN REINFORCED CONCRETE BUILDINGS.

The following very able discussion of this most important subject is presented by Wm. M. Bailey, M. A. M. Soc. C. E., and Chief Engineer, Eastern Concrete Construction Co., Boston, Mass.:

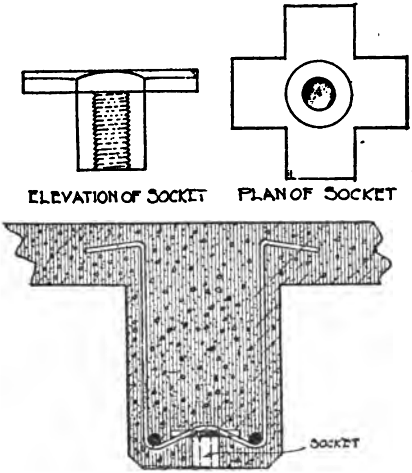


FIG. 79.

The manner of attaching shaft hangers, pipes and machinery to beams and floors in a factory or mill building of the reinforced concrete type, is a detail of considerable importance, so much so in fact, that often if no satisfactory scheme is explained to the owner, he will not consider the use of reinforced concrete beams in his proposed building. In the ordinary mill constructed factory with wooden or steel floor beams, there is no special difficulty in attaching hangers for shafting and pipes, or fastening machinery to the beams and floors in any desired location without any previous plans necessary and at a minimum cost. But in a concrete building the conditions are entirely different, and it is much better to adopt some scheme for the installation of the machinery, piping, etc., before the building is constructed if we wish to obtain the best results in all respects.

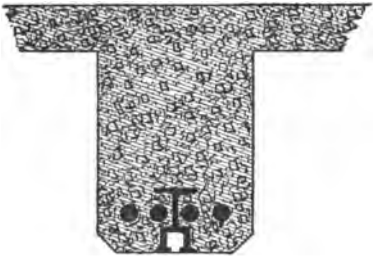


FIG. 80.

Shaft Hanger Supports.—Appliances for conveniently supporting hangers for pipes and shafting, and for supporting motors and other machinery from the beams and ceiling, probably require more previous planning and careful selection than is necessary for other phases of this subject. The method recommended by the engineer and adopted for use with this type of building, should be one designed to best meet all the requirements so far as can be known of the building and its later equipment. It should be inexpensive, compact and simple in form, with not too many separate parts, and as flexible as possible to accommodate future changes in the line of shafting, additions or improved machinery, etc.

One of the most common methods employed, is to attach a hard pine timber to the under side of the concrete beam. This may be in any desired length, to which the shaft hangers are bolted or lag-screwed in their exact position, as is shown in Fig. 75. The timber can be secured to the bottom of the beams by bolts built into the concrete about 4 ft. on centers with a projecting threaded end and nut below, or by the use of sockets, inserts, sleeves and washer nuts imbedded in the concrete, into which the timber pieces are afterwards fastened. If the bolts are used, care should be taken to set and keep them vertical, also have them

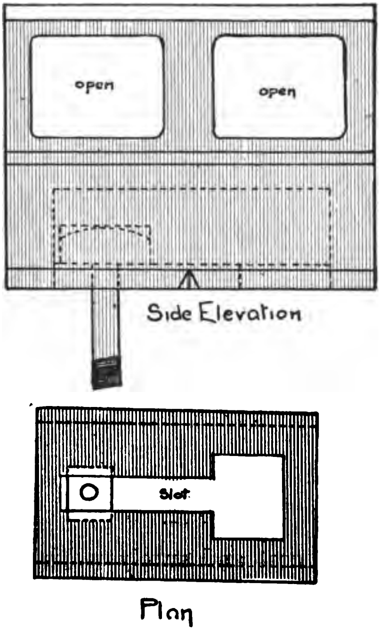


FIG. 81.

thoroughly anchored into the upper part of the beams.

Fig. 76 illustrates the long bolts with saddles and check nuts to hold them in position, which is quite essential. It is seldom that all the bolts projecting from the bottom of the beam are made use of and where not used they present an unsightly appearance, hence sockets or sleeves and washer nuts are often preferable. These can be kept in correct position with less trouble at about the same cost and surely make a neater looking job.

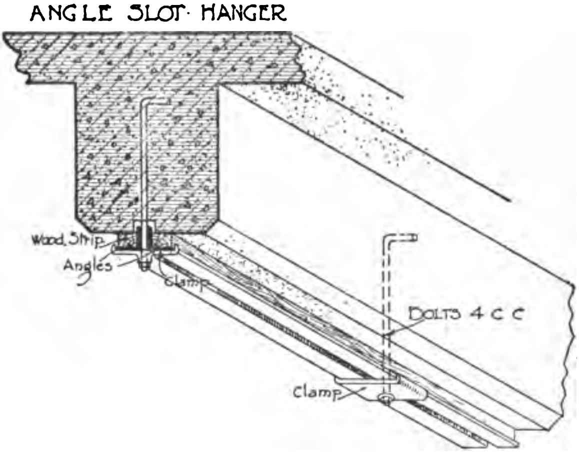


FIG. 82.

In Fig. 77 we have shown the Unit socket, the well known insert for concrete work. This device serves also as a support and spacer for the beam rods, keeping them all properly spaced from each other and from the beam box or centering.

Fig. 78 represents a socket sometimes used. It is made of a short piece of wrought iron pipe tapped out on the inside to receive a lag screw or bolt. Holes are drilled near the top through which a ¼-inch rod or stirrup is placed which should extend well up into the concrete. The socket itself is temporarily held in position before the concrete is placed, by setting it over a wooden plug nailed to the centering. The stirrups are not easily kept in an upright position. For this reason, this socket is not altogether satisfactory.

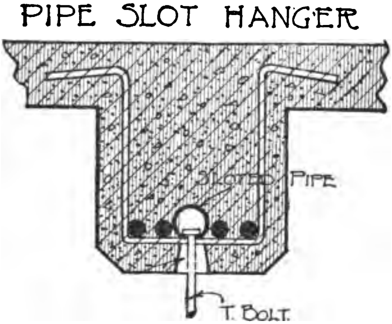


FIG. 83.

The socket shown in Fig. 79 is made of cast, or malleable iron, with a flanged top which serves to anchor it into the concrete. A vertical stirrup should be used with this socket if it is placed in a beam, and is intended to support much load. If used as shown in Fig.

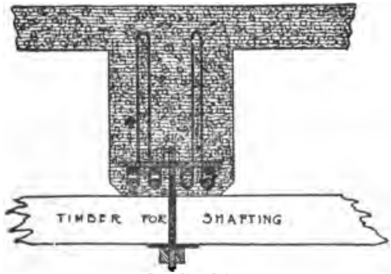


FIG. 84.

79, it also acts as a support for the rods. The Bigelow socket shown in Fig. 80 can be made use of for almost all supports in reinforced concrete work.

It is a malleable casting made in many sizes, can be anchored as securely in the concrete as may be necessary by the use of stirrups, and is more or less adjustable. Another form of adjustable sockets or inserts for use in beams principally is shown in Fig. 81. These castings are made in convenient lengths, and are set on the forms for centering before the concrete is placed.

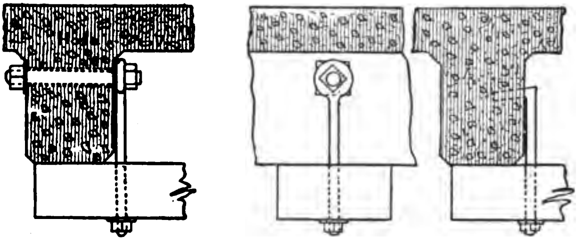


FIG. 85.

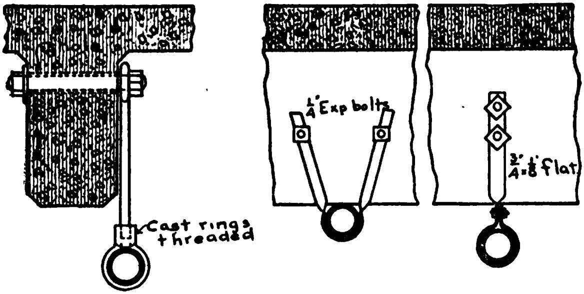
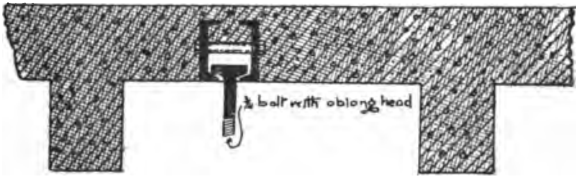


FIG. 86.

The slot in the bottom makes it possible to place hangers, or bolts at any desired location along the length of the insert. Fig. 75 illustrates the use of large washer nuts imbedded in the upper part of the concrete beam, usually about 4 feet on centers. These are temporarily held in position by a thin iron tube resting on the centering, and by a bolt projecting through the hole in the bottom of the beam box. The washer nut is tightened down to the top of the tube which secures it firmly in an upright position. When the beam boxes or centering are taken down, the bolts are easily unscrewed from the nuts, thus leaving a clear passage through the concrete to the nut above, to which can be attached timber, or iron stringer onto which the shaft hangers can be fastened in the ordinary manner.



For slabs between beams

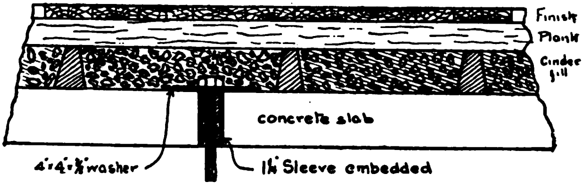


FIG. 87.
METHODS USED FOR SLABS BETWEEN BEAMS.

Figs. 82 and 83 illustrate two other methods used for attaching shaft hangers and machinery to beams. With either of these schemes the shaft hangers or other fittings may be bolted or clamped to the concrete structure without the use of wood timbers, or iron bolsters, if so desired. The method shown in Fig. 82 is one that was adopted by the United Shoe Machinery Company, in its extensive shops at Beverly, Mass.

Another method is suggested in Fig. 84, which may be made a strong and serviceable support for motors and other machinery. Other systems are shown in Figs. 85 and 86. When a bolt is used through the beam near the top, a metal sleeve having a wooden core is concreted around to form the hole. These openings also permit the passage of pipes and wires.

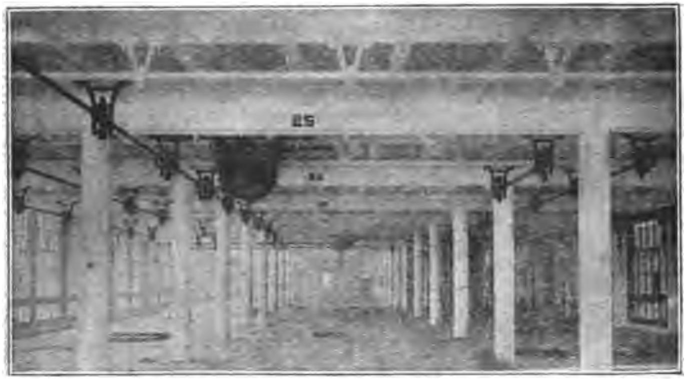


FIG. 88.
GENERAL VIEW OF SHAFTING.

Large holes left in certain parts of beams and girders might cause weakness in the compressive side of the member. In such cases, an iron pipe sleeve of sufficient thickness should be used. It is rather difficult to always locate the holes through the beams during construction, so as to conform exactly with future changes. Therefore, the system using the hooked hanger bolt inside the beam is the most satisfactory. If the holes for the hook bolts are cut with a pneumatic drill, it can be done at a nominal cost.

Wires and Sprinkler Supports.—Blocks of wood are often set in the under side of concrete slabs to which are fastened wires, and small sized pipes, and wire conduits. This is not considered a suitable, or permanent support. It is much better practice to use either expansion bolts, or else locate, and place adjustable sockets, and inserts in the concrete work.

Often the conduits for wires are placed on the centering and imbedded in the concrete slabs. If the pipes are not too large, and are located with due regard to the reinforcing metal, there is no objection to this method. Piping for sprinklers is generally planned before the concrete work is started. It is therefore a simple matter to place inserts, or sockets where needed in the concrete work, to which the pipes can be readily attached later on.

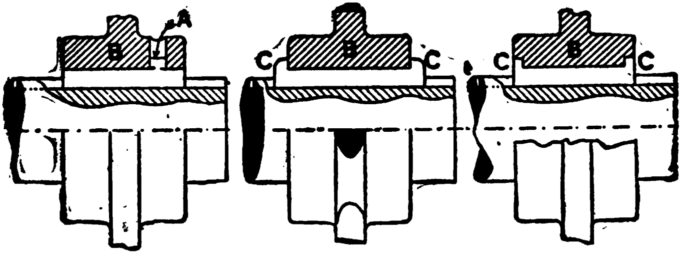


FIG. 89.

SLIDING FEATHER-KEYS.

Machinery Supports.—Large Machines with heavy bases setting on the floor, seldom require any special fastenings. It is usually sufficient to thoroughly grout with Portland cement, the base of the machine to the concrete floor. Small machines may be fastened directly to the concrete by expansion bolts, or similar methods, after the floors are concreted, the cost of drilling for these fastenings being but a small item of expense. Sometimes, however, it may be necessary to bolt through the floor in order to hold certain machines rigidly. If through bolts are required, the holes should, if possible, be located before the floor is concreted; otherwise in cutting these holes through the floor, reinforcing bars are liable to be encountered, which would cause more or less trouble, and might lead to serious difficulty if cut off. In most cases machinery runs better, and lasts longer if placed on solid foundations, and reinforced concrete floors are admirably adapted for machine shops. There are, however, exceptionable cases where machines seem to require more or less "give" on their bases in order to be most satisfactory in working and endurance. This is particularly true with certain textile machinery, and where a cush-

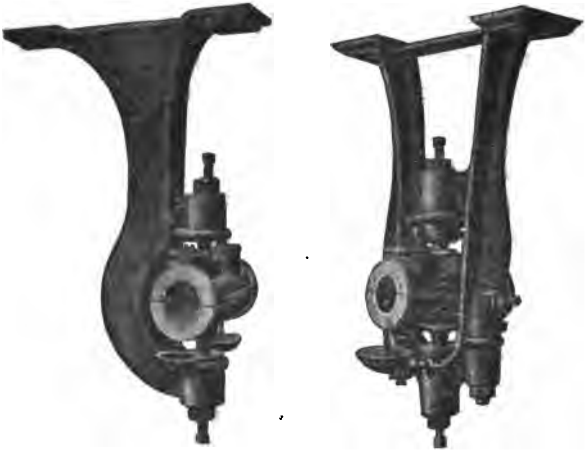


FIG. 90.
MEDART SHAFT HANGERS.

ion is required a wood finished floor can be laid, or wooden timbers or bolsters can be fastened to the concrete flooring. It has also been suggested to use sheet cork or linoleum carpet under the base of machines in order to give elasticity of movement and also lessen the noise of running machinery.

Lining Out, and Leveling for Shafting.—When the bridge-trees have been placed according to the drawing, a line should be stretched either above, or below the timbers, beams or piers, that are to form the support for the shaft, and parallel with the line to which the work is to be done. If when surveying for the foundations of the buildings, permanents stations have been established, it will be an easy matter to work from these stations and transfer a line from them to the bridge-trees, using a station rod, in connection with the transit.

If two station stones have been located directly under the center of the shaft, proceed in the following manner: Set up the transit over one stone, and from this point pick up the center mark on the other station stone. Now have an assistant place a pencil on the edge of one of the bridge-trees, then sight to the bridge-tree with the telescope, directing the movement of the pencil in one or another until it is cut by the cross-hairs of the transit, then have the assistant mark the spot thus located and one point in the shaft line will be located. Proceed in this manner to mark each bridge-tree,. after which the pillow blocks may be located directly to the center marks which have been made. Some workmen prefer to stretch a line from which to work, instead of working directly to transit-given marks on each bridge-tree. In such cases all that is necessary is to mark the first, and the last timber or pier, and then stretch a line fair with these two marks.

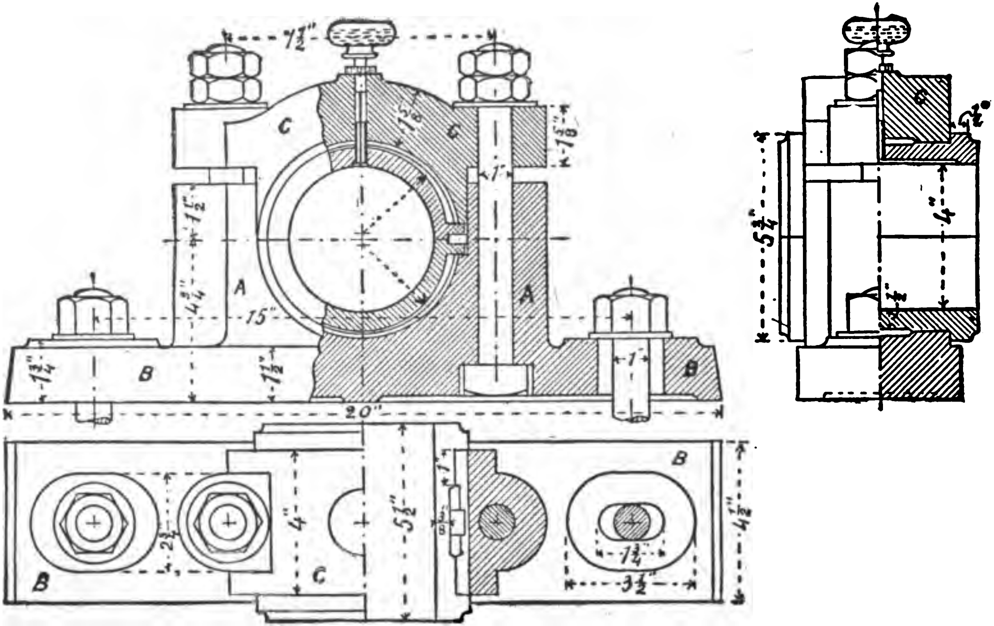


FIG. 90a.
PILLOW BLOCK WITH BRASS BUSHING.

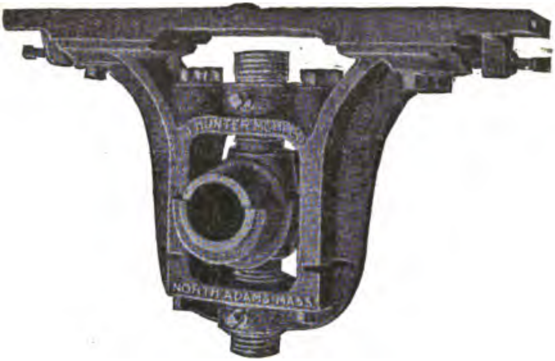

HUNTER MCH. CO.
NORTH ADAMS MASS.

FIG. 91.
HUNTER HEAD SHAFT HANGER.

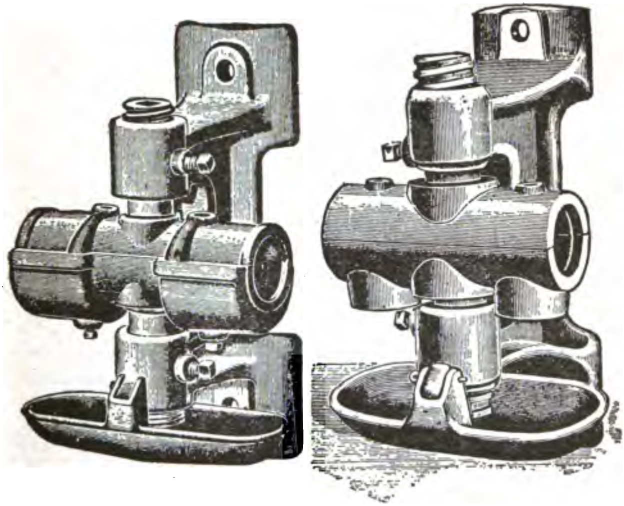


FIG. 92.
GOLDEN'S POST HANGERS.

Various methods of attaching shaft hangers, and brackets to reinforced concrete construction have already been described and illustrated. In cases where it is necessary to put up shafting on old concrete work, either in making changes, or in fitting up buildings not originally intended for mill work, the hangers must be bolted to the concrete beams, necessitating the drilling of numerous holes in the concrete.

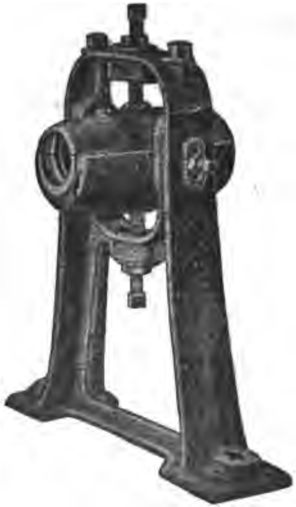


FIG. 93.
HUNTER FLOOR STAND.

Shaft Hangers and Floor Stands. Figs. 90 and 91 show designs for supporting shafting from the ceiling. Fig. 92 is an example of those suited to posts and columns, and Fig. 93 illustrates the Hunter floor stand. The difference in hangers will be found mostly in the methods of adjustment of the boxes, and the facilities for lubrication. Any hanger that may be selected should have the proper distribution of metal to withstand any strain that may in ordinary service be brought to bear, without danger of breaking or springing. There must also be suitable adjustment in all directions for aligning the shaft when it is being placed in position. The distance between hangers will of course vary with the size of the shafting, and the work that it is expected to do, as for instance the number of pulleys to be carried, and the amount of power that is to be transmitted by it, the arrangement of the machines to which the power is to be transmitted, and the speed of the shaft.

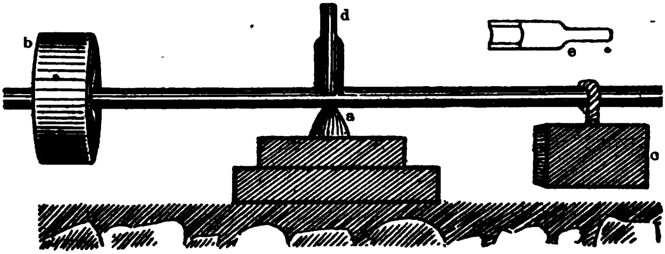


FIG. 94.

All shafting must be true to size and straight if good results are to be obtained, and in order that pulleys may be properly fitted, and when in place, will run true and accurately. The satisfaction that will be obtained from the use of shafting depends, to a large extent, on the arrangement of the machines which are driven from it. As a rule, shafting requires to be straightened after the key seats have been cut, and the couplings have been fitted on.

Fig. 94 illustrates one method that may be used in the absence of a wheel press. The bend is placed on the solid metal bearing, a, with the hollow side of the shaft uppermost, although it appears in the cut to be bent downwards due to the fact that one end has been loaded with the pulley b, and other weights, while the other end is weighted as shown at c. Enough weight should be placed to bring the ends of the shaft about three inches below a horizontal line through the bearing a, and the ends of the shaft when not loaded. With this temporary strain on the shaft, it may be straightened by using a sledge hammer and the tool e, applied as shown at d. The end of tool e should be made to fit the circle of the shaft, in order to prevent the shaft being flattened or otherwise distorted by the blows from the sledge hammer. In this manner the "short" or hollow side of the shaft may be elongated, but in order to determine just how much of this treatment is necessary, the shaft should frequently be unloaded and revolved between centers, or other supports at the ends. By repeated and careful work in peening, the shaft may be accurately straightened by this method.

Setting Up and Babbitting Journal Bearings.—While on the subject of shaft erection, a space will be devoted to the above very important matter. First, the pillow blocks should have solid bearings and rest firmly on their support. If in cutting in, the mistake has been made of cutting too deep for the bearing, the best remedy is to make the cut deep enough to allow of the placing of piece of board of standard thickness ½, ¾ or ⅞ inches as a base for the pillow block to rest upon. Before bolting it down, the pillow block casting should be rubbed back and forth on its bearing to ascertain whether or not there are any high places on the surfaces of the bearing, or the face of the casting. The bearing should be rubbed with blue or red chalk which will, when the two surfaces are brought together, clearly indicate the high places.

TABLE 19—U. S. STANDARD BOLTS AND NUTS.

DIA. OF SCW	THDS PER INCH	DIA AT ROOT OF THD	AREA IN SQ. IN AT ROOT OF THD	HEX. HEAD BOLTS					HEX. NUTS			SQUARE HEAD BOLTS					SQUARE NUTS			TAP DRILL SIZE	TENSION AT 10,000 LB. PER SQ. INCH	SHEAR AT 8,000 LB. PER SQ. INCH.
				FIN. HEAD		RGH. HEAD		THICKNESS OF HD	FINISHED			FIN. HEAD		RGH. HEAD		THICKNESS OF HD	FINISHED					
				SHORT DIA. OF HEAD	LONG DIA. OF HEAD	SHORT DIA. OF HEAD	LONG DIA. OF HEAD		SHORT DIA.	LONG DIA.	THICKNESS	SHORT DIA. OF HEAD	LONG DIA. OF HEAD	SHORT DIA. OF HEAD	LONG DIA. OF HEAD		SHORT DIA.	LONG DIA.	THICKNESS			
1/4	20	.185	.027	7/16	15/32	1/2	9/16	1/4	7/16	15/32	3/16	7/16	5/8	1/2	23/32	1/4	7/16	5/8	3/16	3/16	269	392
5/16	18	.240	.045	17/32	19/32	19/32	11/16	19/64	17/32	19/32	1/4	17/32	3/4	19/32	27/32	19/64	17/32	3/4	1/4	1/4	454	615
3/8	16	.294	.068	5/8	11/16	11/16	25/32	11/32	5/8	11/16	5/16	5/8	7/8	11/16	31/32	11/32	5/8	7/8	5/16	5/16	679	884
1/2	13	.400	.126	13/16	29/32	7/8	1	7/16	13/16	29/32	7/16	13/16	1 5/32	7/8	1 1/4	7/16	13/16	1 5/32	7/16	27/64	1257	1571
5/8	11	.507	.202	1	1 1/8	1 1/16	1 7/32	17/32	1	1 1/8	9/16	1	1 13/32	1 1/16	1 1/2	17/32	1	1 13/32	9/16	17/32	2019	2454
3/4	10	.620	.302	1 3/16	1 11/32	1 1/4	1 7/16	5/8	1 3/16	1 11/32	11/16	1 3/16	1 21/32	1 1/4	1 3/4	5/8	1 3/16	1 21/32	11/16	41/64	3019	3534
7/8	9	.731	.420	1 3/8	1 9/16	1 7/16	1 21/32	23/32	1 3/8	1 9/16	13/16	1 3/8	1 15/16	1 7/16	2 1/32	23/32	1 3/8	1 15/16	13/16	3/4	4197	4810
1	8	.837	.550	1 9/16	1 25/32	1 5/8	1 7/8	13/16	1 9/16	1 25/32	15/16	1 9/16	2 3/16	1 5/8	2 9/32	13/16	1 9/16	2 3/16	15/16	55/64	5502	6283
1 1/8	7	.940	.694	1 3/4	2	1 13/16	2 3/32	29/32	1 3/4	2	1 1/16	1 3/4	2 15/32	1 13/16	2 9/16	29/32	1 3/4	2 15/32	1 1/16	31/32	6940	7952
1 1/4	7	1.065	.893	1 15/16	2 7/32	2	2 5/16	1	1 15/16	2 7/32	1 3/16	1 15/16	2 23/32	2	2 13/16	1	1 15/16	2 23/32	1 3/16	1 3/32	8908	9817
1 3/8	6	1.160	1.057	2 1/8	2 7/16	2 3/16	2 17/32	1 3/32	2 1/8	2 7/16	1 5/16	2 1/8	3	2 3/16	3 3/32	1 3/32	2 1/8	3	1 5/16	1 11/64	10541	11872
1 1/2	6	1.284	1295	2 5/16	2 21/32	2 3/8	2 3/4	1 3/16	2 5/16	2 21/32	1 7/16	2 5/16	3 9/32	2 3/8	3 3/8	1 3/16	2 5/16	3 9/32	1 7/16	1 5/16	12948	14136
1 5/8	5 1/2	1.389	1.515	2 1/2	2 7/8	2 9/16	2 31/32	1 9/32	2 1/2	2 7/8	1 9/16	2 1/2	3 17/32	2 9/16	3 5/8	1 9/32	2 1/2	3 17/32	1 9/16	1 25/64	15149	16584
1 3/4	5.	1.491	1.744	2 11/16	3 3/32	2 3/4	3 3/16	1 3/8	2 11/16	3 3/32	1 11/16	2 11/16	3 25/32	2 3/4	3 7/8	1 3/8	2 11/16	3 25/32	1 11/16	1 17/32	17460	19242
1 7/8	5	1.615	2.048	2 7/8	3 5/16	2 15/16	3 13/32	1 15/32	2 7/8	3 5/16	1 13/16	2 7/8	4 1/32	2 15/16	4 1/8	1 15/32	2 7/8	4 1/32	1 13/16	1 5/8	20490	22080
2	4 1/2	1.712	2.302	3 1/16	3 17/32	3 1/8	3 5/8	1 9/16	3 1/16	3 17/32	1 15/16	3 1/16	4 11/32	3 1/8	4 7/16	1 9/16	3 1/16	4 11/32	1 15/16	1 3/4	23020	25133
2 1/4	4 1/2	1.962	3.023	3 7/16	3 31/32	3 1/2	4 1/16	1 3/4	3 7/16	3 31/32	2 3/16	3 7/16	4 27/32	3 1/2	4 15/16	1 3/4	3 7/16	4 27/32	2 3/16	2	30234	31809
2 1/2	4	2.176	3.715	3 13/16	4 13/32	3 7/8	4 1/2	1 15/16	3 13/16	4 13/32	2 7/16	3 13/16	5 13/32	3 7/8	5 1/2	1 15/16	3 13/16	5 13/32	2 7/16	2 7/32	37189	39270

In boring or drilling the bolt holes, care should be exercised that they be bored fair with the holes in the casting.

When preparing journal bearings, for babbitting, two very important points should not be overlooked, viz: to see that they are clean, and dry. All dirt, and dust should be thoroughly cleaned from the cavities in the castings, after which the casting should be dried by being placed over a forge fire, or if too heavy to be handled in this way, it may be dried by pouring a small quantity of gasoline into the spaces to be babbitted and then setting the fluid on fire. This is in fact a better plan than drying over a fire for the reason that the gasoline will find its way into all the holes and small cavities, thus insuring a thorough drying. A very small quantity of water left in the space to be babbitted will cause lots of trouble when the hot metal is poured in. If the surfaces which are to receive the babbitt metal are heated to a temperature slightly lower than the melting point of the soft metal, the conditions for pouring the bearing may be considered as ideal.

Table 20 gives the constituents of most of the prominent bearing metals as analyzed at the Pennsylvania Railroad laboratory at Altoona, Penn.

The practice of lining journal-boxes with a metal that is sufficiently fusible to be melted in a common ladle is not always so much for the purpose of securing antifriction properties as for the convenience and cheapness of forming a perfect bearing in line with the shaft without the necessity of boring them. Boxes that are bored, no matter how accurate, require great care in fitting and attaching them to the frame or other parts of a machine.

TABLE 20.

BEARING-METAL ALLOYS.

Trade Name of Alloy	Copper	Tin	Lead	Zinc	Antimo'y	Iron.
Camelia metal	70 20	4 22	14 75	10 20		0.55
Anti-friction metal	1.60	98 13				trace
White metal			87 92		12 08	
Car-brass lining		trace	84 87		15 10	
Salgee anti-friction	4 01	9 91	1 15	85.57		
[1] Graphite bearing-metal		14 38	67 73		16.73	
Antimonial lead			80 69		18 83	
[2] Carbon bronze	75 47	9 72	14 57			
[3] Cornish bronze	77 83	9 60	12 40			
Delta metal	92 39	2 37	5 10			0 07
[4] Magnolia metal			83 55	trace	16 45	trace
American Anti-friction metal			77.44	0 98	19 60	0.65
Tobin bronze	59 00	2 16	0 31	38.40		0.11
Graney bronze	75 80	9 20	15 06			
Damascas bronze	76 41	10 60	12 52			
[5] Manganese bronze	90 52	9 58				
[6] Ajax metal	81 24	10 98	7 27			
Anti-friction metal			88.32		11 93	
Harrington bronze	55 73	0 97		42 76		0.68
Car-box metal			84 33	trace	6 03	
[7] Phosphor bronze	79 17	10 22	10 61			
[8] P.R R "B" metal	76 80	8	15 00			
[9] Babbitt metal (parts)	1	50			5	
Babbitt metal light	1 8	89 3			8 9	
Babbitt metal best	3 7	88 9			7 4	
[10] Babbitt (parts) metal best	4	96			8.	
Another "babbitt"	1 5	45 5	40 00		16	
Brittania	1	85 7		2 9		
Brittania		81 9		1 9	16.2	
Brittania	2	81		1	16	
Brittania	4	70 5			25.5	
Brittania	10	22		6	62	
[11] Plate pewter	1 8	89 3			7 1	
White metal bearings on German locomotives	5	85			10	
French white metal h'd	10		65		25	
French white medium	5 5	83 3			11 2	
French white soft		10	70		20.	
French white very soft		12	80		8.	
English.—"Parsons"	2	86	2	27	1	
English —"Richards"	4 5	70	10 5		15.	
English —"Babbitt"	3 5	55	23 5		18	
English —"Fenton's"	5	16		79		
English —"French Navy"	7	7 5	7	87 5		
English:—"German Navy"	7 5	85			7 5	
Type metal, soft, to			83		17	
Type metal, hard			80		20	
Ornamental castings			66 63		33 33	
[12] Pattern metal		10	10		2	

---

[1] Contains no Graphite. [2] Contains a possible trace of carbon. [3] Trace of zinc, Iron and Phosphorus. [4] Dr. H. C. Torrey says this analysis is erroneous and that Magnolia metal always contains tin. [5] Contains no Manganese. [6] Phosphorus or Arsenic 0.37. [7] Phosphorous 0.94. [8] Phosphorous 0.20. [9] (In parts by weight). [10] (In parts by weight). [11] Bismuth 1.8. [12] Bismuth 6. Brass 8.

It is not good practice, however, to use the shaft for the purpose of casting the bearings, especially if the shaft be steel, for the reason that the hot metal is apt to spring it; the better plan is to use a mandrel of the same size or a trifle larger for this purpose. For slow-running journals, where the load is moderate, almost any metal that may be conveniently melted, and will run free will answer the purpose. For wearing properties, with a moderate speed, there is probably nothing superior to pure zinc, but when not combined with some other metal it shrinks so much in cooling that it cannot be held firmly in the recess, and soon works loose; and it lacks those anti-friction properties which are necessary in order to stand high speed.

For line-shafting, and all work where the speed is not over 300 or 400 R. P. M., an alloy of 8 parts zinc and 2 parts block-tin will not only wear longer than any composition of this class, but will successfully resist the force of a heavy load. The tin counteracts the shrinkage, so that the metal, if not overheated, will firmly adhere to the box until it is worn out. But this mixture does not possess sufficient anti-friction properties to warrant its use in fast-running journals.

Among all the soft metals in use, there are none that possess greater anti-friction properties than pure lead; but lead alone is impracticable, for it is so soft that it cannot be retained in the recess. But when by any process lead can be sufficiently hardened to be retained in the boxes without materially injuring its anti-friction properties, there is no metal that will wear longer in light, fast-running journals. With most of the best and most popular anti-friction metals in use and sold under the name of Babbitt metal, the basis is lead.

Lead and antimony have the property of combining with each other in all proportions without impairing the anti-friction properties of either. The antimony hardens the lead, and when mixed in the proportion of 80 parts lead by weight with 20 parts antimony, no other known composition of metals possesses greater anti-friction or wearing properties, or will stand a higher speed without heat or abrasion. It runs free in its melted state, has no shrinkage, and is better adapted to light high-speeded machinery than any other known metal. Care, however, should be exercised in using it, and it should never be heated beyond a temperature that will scorch a dry pine stick.

Many different compositions are sold under the name of Babbitt metal. Some are good, but more are worthless; while but very little genuine Babbitt metal is sold that is made strictly according to the original formula. Most of the metals sold under that name are the refuse of type-foundries and other smelting-works, melted and cast into fancy ingots with special brands, and sold under the name of Babbitt metal.

It is difficult at the present time to determine the exact formulas used by the original Babbitt, the inventor of the recessed box, as a number of different formulas are given for that composition. Tin, copper, and antimony were the ingredients, and from the best sources of information the original proportions were as follows:

		Another writer gives:
50 parts tin	= 89.3%	83.3%
2 parts copper	= 3.6%	8.3%
4 parts antimony	= 7.1%	8.3%

The copper was first melted, and the antimony added first and then about ten or fifteen pounds of tin, the whole kept at a dull-red heat and constantly stirred until the metals were thoroughly incorporated, after which the balance of the tin was added, and after being thoroughly stirred again, it was then cast into ingots. When the copper is thoroughly melted, and before the antimony is added, a handful of powdered charcoal should be thrown into the crucible to form a flux, in order to exclude the air and prevent the antimony from vaporizing; otherwise much of it will escape in the form of a vapor and consequently be wasted. This metal, when carefully prepared, is probably one of the best metals in use for lining boxes that are subjected to a heavy weight and wear; but for light fast-running journals, the copper renders it more susceptible to friction, and it is more liable to heat than the metal composed of lead and antimony in the proportions just given.

FIG. 95.

Preparations for Pouring Babbitt Metal.—Provide two pieces of shafting for mandrels of a length somewhat longer than the bearing or box to be babbitted. One of these mandrels is to be turned to a size about $\frac{1}{32}$ inch larger than the diameter of the shaft, and this is to be used for shaping the babbitt in pouring.

The other mandrel should be of the same diameter as the shaft which is to run in the bearing, and this mandrel is for use when scraping the babbitt metal to a bearing.

If it is inconvenient to procure a piece of shafting larger in diameter than the journal, the difficulty may be gotten over by placing a sheet of strong smooth paper around the mandrel that is of the same diameter as the shaft and then pour the metal around it.

The thickness of the paper around the mandrel, and the contraction of the soft metal when cooled will serve to relieve the journal from being pinched by the babbitt metal, which is liable to occur when the metal is poured directly around the journal. Two thin pieces of wood liners, one for each side of the box, should also be prepared, and they should be of about the thickness of the permanent liners to be used. One edge of these wooden liners is to be notched so as to allow the metal when poured from above to fill the lower box.

These liners are placed between the cap and bottom half of the box with the shaft in a central position and the parts of the box clamped together. In case the box is secured together by means of bolts the outer edges of the liners mentioned are slotted to fit these bolts. Small wooden blocks, cut taper, are placed in each end of the box to maintain the shaft in a central position. The box is placed horizontally in a vise and the shaft centered by means of calipers, the shaft being moved by moving the wedges in or out as needed. After the shaft is correctly located, the ends of the box are stopped by means of putty, or any other convenient method which will prevent the hot babbitt from flowing out. A putty funnel is also built up about the hole into which the hot babbitt is to be poured.

The babbitt is heated in any convenient way until it is fluid. Care should be taken, however, that it does not get too hot, for if it does its efficiency will be diminished. A good method of testing is to dip a piece of soft pine into the hot metal, and when it is just hot enough to char the wood the proper temperature has been reached. It is then poured into the box through the funnel already prepared, as rapidly as possible, as it quickly sets and if this condition occurs the box will be spoiled and must be cleaned and repoured.

After the metal has set the sprue in the funnel is broken from the cap, the cap removed from the lower part of the box and the shafting removed. The babbitt should have formed a reasonably smooth surface, and be free from cold shots. In cold weather it is sometimes advisable to warm the shaft used as a mandrel.

When the cap and the box are poured separately, the melted babbitt metal is usually poured into one side of the bearing and rises along the other side until the cavity is full.

When pouring a bearing of this character, move the ladle from one end of the bearing to the other, causing the stream to traverse along the slot, thereby keeping the babbitt at approximately the same temperature in all parts of the bearing. When two ladles are used to pour from, it will be sufficient to pour into diagonally opposite corners of the bearing, and if both streams be poured quickly, the cavity will be filled before the metal becomes too cold to flow to the corners.

Oil-channels may be formed in the lining by means of a small, hard-spun cord wound around the mandrel in the direction that the shaft is to run. The cord should be rubbed smooth with putty before being placed upon the mandrel.

Pouring Thin Solid Boxes.—It was stated above that the temperature of the alloy when poured should be just high enough to slightly char a soft-wood stick. This is a general rule, and by following it the millwright will not go far astray on ordinary work. But there are exceptions: when thin solid boxes must be poured, and there is a long narrow or thin space to be filled with soft metal, then it is sometimes necessary to pour the metal hotter than described.

In such cases, the metals to come in contact with the soft metal should always be heated as much as possible before pouring in the lining.

When a ladleful of metal fails to fill a box, and another ladleful is not at hand, do not try to fill the box by pouring hot metal on top of the cold, for it is very seldom that such a course results in a satisfactory bearing. Hot babbitt will not unite with or weld itself to cold babbitt by simply pouring one on top of the other; therefore, should a man fail to fill a bearing when pouring in the lining, it is decidedly the best practice to take down and chip out the partially filled bearing, set it up again, and pour from a supply sufficient to fill the bearing.

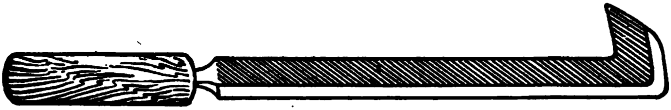


FIG. 96.

Pouring Cold Boxes.—When babbitt or other lining metal is poured into a cold casting, the lining becomes chilled before the iron casting becomes heated; therefore when the soft lining finally cools it shrinks away from the casting and becomes loose, and must be peened to make it tight in the journal bearing casting. Heating the box before pouring in the soft metal lining will prevent this looseness, as the iron casting then shrinks with the soft metal lining on cooling and holds it fast when cold.

There is another way of fastening a lining tightly into a cold casting and that is by using one of the antimony alloys as a lining. Antimony, when alloyed with lead and with some other soft metals, loses the power of shrinking during the freezing process, and like water, expands and fills the casting so tightly that there is no rattle or looseness. Therefore, when forced to line a cold bearing, use an antimony alloy. It is for this purpose that the antimony alloy is used for casting printing type. The expansion of the alloy during the instant of solidifying or freezing causes the metal to expand into every corner of the mold, thereby securing the extreme sharpness necessary in printing type. The same is true with the antimony alloy in the journal bearing.

Scraping a Bearing.—In order to obtain a good and reliable journal bearing the babbitt should, after cooling, be scraped and accurately fitted to the journal that is to run in it, and for this work the mandrel previously referred to which has been turned to the same size as the shaft, is now covered with ultramarine, and placed in the lower half of the box, and revolved. Upon removing it the high spots in the babbitt will be plainly marked.

There may be only two or three spots so marked at the first trial, but these indicate that the shaft touches the bearing only at those spots, and the metal should be removed at those points by scraping, after which the mandrel should again be placed in the bearing and revolved. This time the shaft may touch in four or five places, as indicated by the markings. This process should be repeated until the bearing spots appear practically all over the surface of the babbitted box, when it may be assumed that the job is complete.

Plumbago, or red lead mixed with oil may be used on the mandrel for the marking in the absence of ultramarine, and the mandrel should be rubbed with it each time it is tried in the bearing. The scraping is done with a tool, the usual form of which is shown in Fig. 95. In order that the tool may be held with ease in the hands, it should be longer than the length of the bearing, long enough in fact so that one end will project beyond one end of the box when the work is being done. The scraper shown in Fig. 95 may be forged from an old worn out half round file, although it is better to make it from a piece of tool steel. The method of using this scraper on a newly babbitted bearing is as follows: holding the back of the tool between the thumb and fingers of each hand, bring the cutting edge flatly against the bearing metal and carefully shave off all the high spots and lumps which are marked by the turning of the mandrel in the bearing. For small bearings a smaller tool should be used. Another design of scraper is shown in Fig. 96, but not nearly so accurate work can be done with this tool as can be done with the tool shown in Fig. 95. A worn out flat file of ordinary size may be utilized to good advantage in making the scraper shown in Fig. 96. The end of the file is first beveled at an angle of 45 degrees and the edge thus formed on the end is rounded as shown in the cut, after which another heat is taken and the bend put in at an angle of 75 or 80 degrees.

The tool is then ground to a sharp edge. The method of using the scraper shown in Fig. 96 is just the reverse of that described in connection with Fig. 95, which was crosswise of the bearing, Fig. 96 is to be used lengthwise. Great care, and considerable skill must be exercised in the use of scraper shown in Fig. 96, as there is danger of cutting too deep if the handle is not held correctly.

Ball and Roller Bearings. The ball bearing is most suitable for high speeds and light loads. For instance, on the spindles of wood carving machines, some make as much as 30,000 revolutions per minute. When shaft journals are mounted upon ball bearings the total bearing surface presented by even a considerable number of balls is very small comparatively speaking, and the consequence is that a large amount of wear occurs in a short length of time. For this reason the life of a ball bearing is limited to a very short period compared with that of the ordinary babbitted bearing. Neither have ball bearings been found satisfactory for thrust blocks, much better results having been obtained from coned rollers. In some cases a combined system of rollers and balls has been used with satisfactory results. Ball and roller bearings serve to greatly decrease the friction of shaft journals, besides requiring very little if any lubrication. When, in order to increase the bearing surface, the length of a ball bearing is increased it practically becomes a roller bearing. A comparison of the co-efficients of friction of well lubricated journal bearings and ball or roller bearings follows.

Journal bearings—co-efficient = 0.09 to 0.15
Roller bearings—co-efficient = 0.01 to 0.03

High grade roller-bearings are made by several concerns, but when bearings of this type are to be attached to a shaft, unless great care is taken that the race, or shell for the rolls is in exact line with the shaft, there will be unequal wear which will more than counteract the saving in power by reduced friction.

When roller bearings are used, one of the principal requirements is that the shaft journal surface shall be perfectly true and smooth, and as nearly a perfect cylinder as it is possible to make it. Unless these requirements are fulfilled there will be no end of trouble, and the life of the bearing will be short. Roller, and ball bearings also require care and attention in order to give satisfactory results in operation. They should be cleaned at frequent intervals, and they also require a certain amount of lubrication, although not so much as the ordinary babbitted bearing does. The balls should be of the largest possible diameter that the space at disposal will admit of, also any one ball should be capable of carrying the total load upon the bearing. Until the crushing strength of the balls is neared, the frictional resistance is proportional to the load. The resistance is independent of the number of balls, and of the speed.

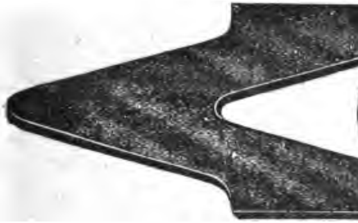


FIG. 97.
SHEET STEEL BLANK.

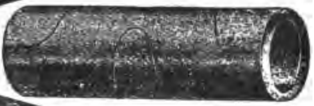


FIG. 97a.
HOLLOW CYLINDER ROLLER.

Pin Bearings are a form of roller-bearing used to a considerable extent on push cars in use in factories and mills. These cars are intended to carry heavy loads, and are pushed about by hand on the floor, or on a track as needed. The pin bearings are made of plain short pieces of cold-rolled steel rod, cut to length, and interposed between the journal and housing of the bearing. Three important factors enter in the design of roller and pin bearings, viz., the diameter of the journal, the diameter of the rollers, and the number of rollers. In case the millwright should be called upon to design a roller-bearing, or ball-bearing to fill a given housing upon a shaft whose diameter is known, he may be guided by the following rule and formula: Rule: the diameter of a circle through the center of the balls equals the diameter of a ball divided by the sine of the angle occupied by one-half of the ball.

Kynoch Roller Bearings. The rollers are made from a special quality of sheet steel, and are formed by rolling a blank of the shape shown by Fig. 97 into a hollow cylinder as seen at Fig. 97a. The advantage of this construction for a roller is that the fibre, or grain of the steel is always in the direction of rotation which gives them an amount of elasticity that solid soft rollers do not possess. The rollers are not hardened and therefore do not laminate the shaft. The rollers should always be kept parallel to the axis of the shaft. This is done by

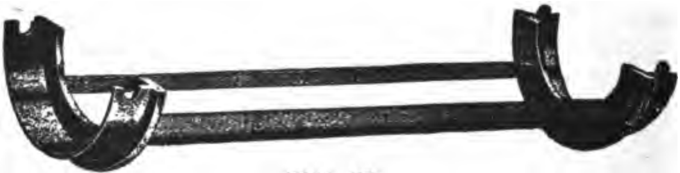


FIG. 97b.

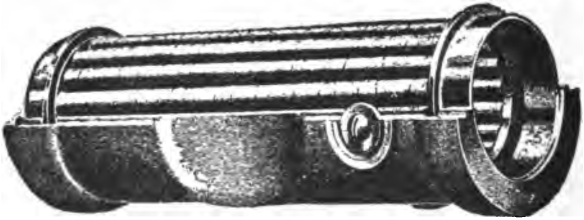


FIG. 98.
KYNOCH ROLLER BEARING WITH TOP CAP REMOVED.

enclosing them in a guiding cage shown in Fig. 97b which is divided into three equal spaces.

Fig. 98a shows a complete Kynoch roller bearing and pillow block. From actual tests made it has been found, that the coefficient of friction between a shaft journal and these roller bearings is only .0115 for repose, and .0082 at the normal speed of rotation. This is claimed to be about one-eighth of the co-efficient of friction (.07) for the same shaft when run in plain babbitted bearings with ordinary lubrication. There is no doubt that there is a saving in labor, oil, and in the transmission of power by the use of these bearings.

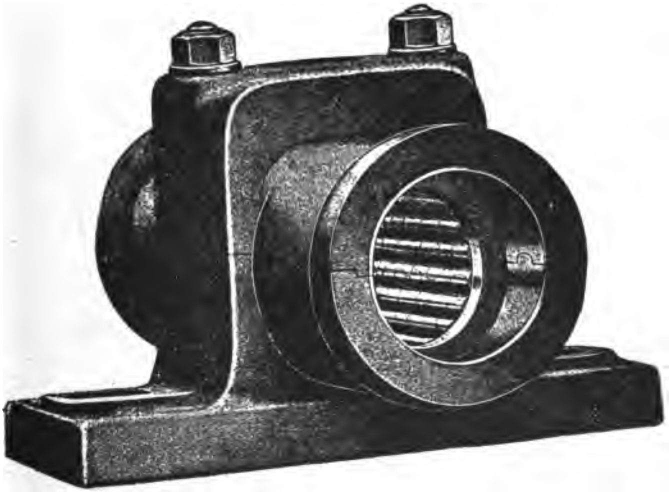


FIG. 98a.
COMPLETE KYNOCH ROLLER BEARING AND PLUMMER BLOCK.

Ball Bearings. It has been explained, that the work lost in friction must, in all cases, be proportional to the distance through which the friction acts. Hence, if by any convenient means we can reduce this distance (without at the same time increasing the other quantities which make up the total losses due to friction), and introduce the distinctly anti-friction qualities of rolling resistance, instead of the more wasteful sliding resistance, we shall save power in driving machines as well as wear and tear in their bearings.

With ball bearings, rolling friction takes the place of the sliding friction in ordinary bearings. Consequently, the chief point to be attained in designing ball bearings is to get true rolling motion of the balls without any grinding action.

For instance, the wheels of a dog-cart are made large in diameter, that a horse may the more easily pull the vehicle over the stones and the other obstacles to be met with on ordinary roads; but the axle bearings are made small with the object of minimizing the distance through which their friction acts, for a given length traversed by the vehicle. And, axle friction can be still further reduced by the use of ball bearings, as has been done in the cases of bicycles, tricycles, and motor cars.

TABLE 21

POWER THAT STEEL SHAFTING WILL TRANSMIT AT VARIOUS SPEEDS.

From *The Practical Engineer*, September 2, 1892. By A. G. Brown, M.E.

Revs. per Minute.	DIAMETERS OF SHAFTS IN INCHES.													
	1½	1¾	2	2¼	2½	3	3½	4	5	6	7	8	9	10
	HORSE-POWERS THEY WILL TRANSMIT.													
50	3·3	5·3	8·0	10·9	15·6	27	43	64	125	216	343	512	729	1000
60	4·0	6·4	9·6	13·1	18·8	32	51	77	150	259	412	614	875	1200
70	4·7	7·5	11·2	15·2	21·9	38	60	89	175	302	480	717	1021	1400
80	5·4	8·5	12·8	17·4	25·0	43	69	102	200	346	549	819	1166	1600
90	6·0	9·6	14·4	19·6	28·1	49	77	115	225	389	617	922	1312	1800
100	6·7	10·7	16·0	21·8	31·2	54	86	128	250	432	686	1024	1458	2000
110	7·4	11·8	17·6	23·9	34·4	59	94	141	275	475	755	1126	1604	2200
120	8·1	12·9	19·2	26·1	37·5	65	103	154	300	518	823	1229	1750	2400
130	8·7	13·9	20·8	28·3	40·6	70	111	166	325	562	892	1331	1895	2600
140	9·4	15·0	22·4	30·5	43·8	76	120	179	350	605	960	1434	2041	2800
150	10·1	16·1	24·0	32·6	46·9	81	129	192	375	648	1029	1536	2187	3000
160	10·8	17·1	25·6	34·8	50·0	86	137	205	400	691	1097	1638	2333	3200
170	11·5	18·2	27·2	37·0	53·1	92	146	218	425	734	1166	1741	2479	3400
180	12·2	19·3	28·8	39·2	56·3	97	154	230	450	778	1235	1843	2624	3600
190	12·8	20·4	30·4	41·3	59·4	103	163	243	475	821	1303	1945	2770	3800
200	13·5	21·4	32·0	43·5	62·5	108	172	256	500	864	1372	2048	2916	4000
225	15·2	24·1	36·6	49·0	70·3	122	193	288	563	972	1543	2304	3280	4500
250	16·9	26·8	40·0	54·4	73·1	135	214	320	625	1080	1715	2560	3645	5000
275	18·6	29·5	44·0	59·8	85·9	149	236	352	688	1188	1886	2816	4009	5500
300	20·3	32·2	48·0	65·3	93·7	162	257	384	750	1296	2058	3072	4374	6000
325	21·9	34·8	52·0	70·7	101·6	176	279	416	813	1404	2229	3328	4739	6500
350	23·6	37·5	56·0	76·2	109·4	189	300	448	875	1512	2401	3584	5103	7000
375	25·3	40·2	60·0	81·6	117·2	203	322	480	938	1620	2572	3840	5468	7500
400	27·0	42·9	64·0	87·0	125·0	216	343	512	1000	1728	2744	4096	5832	8000
425	28·7	45·6	68·0	92·5	132·8	230	364	544	1063	1836	2915	4352	6197	8500
450	30·4	48·2	72·0	97·9	140·6	243	386	576	1125	1944	3087	4608	6562	9000
475	32·1	50·9	71·0	103·4	148·4	257	407	603	1188	2052	3258	4864	6926	9500
500	33·7	53·6	80·0	108·8	156·2	270	429	640	1250	2160	3430	5120	7290	10000

For power of wrought-iron shafts take 70 per cent, of steel shafts of the same size.

Size of Bearings. For ordinary shafting the length is one and a-half times the diameter. To avoid heating, the pressure per square inch must not exceed a certain limit, which limit depends on the speed as well as on the coefficient of friction. There is no reliable definite ratio between the speed and limit of pressure beyond that which is fixed by experience for the different conditions. For ordinary shafting the pressure per square inch of rubbing surface multiplied by the surface velocity in feet per minute should not exceed 20,000. Where the direction of load varies, as in crank pins, the limit may be double of that where it is constantly in one direction, and where the direction of rotation varies as well as the load as in cross-heads, a still greater limit is permissible. Table 24 gives some values of the limit of pressure for different cases, the projected or axial area of the bearing being taken.

Set Collars. There should be two set collars on each shaft. The better plan is to locate these collars at opposite ends of the same bearing, or "box" as it is usually termed, as there will then be but one bearing to watch in case there is too much end play in the shaft. Another argument in favor of this plan is, that in the case of very long shafts, variations in temperature cause such variations in the length of the shaft as to render the proper adjustment of the end play impossible when the set collars are located at the ends of the shaft, whereas if placed at opposite ends of the same bearing, variation in temperature will cut a very small figure.

Set collars should never be so placed upon a shaft as to prevent a certain amount of end play of the shaft. This applies especially to line shafting, and all other cases where a slight end motion can be allowed. There are, of course, many machines, such as circular saws and similar machines, where end motion cannot be permitted.

Electric generators and motors will run much better in every respect if they be given from 1/16 to 3/16 inch end motion. The amount of end motion to be given to shafting and machines depends upon conditions and the kind of machinery, but the clearances given above will generally apply.

Pressure Suitable for a Given Velocity of Journal.—When the product of the pressure multiplied by the velocity of a journal exceeds a certain limit there is danger of heating. The working pressure should therefore be considerably less than that at which heating may commence.

The cool-running pressure, and that under which bearings may become heated, also the pressure generally producing very hot bearings, may be determined by the following rule:

Let $P =$ the maximum pressure in pounds per square inch permissible on the horizontal bearing-surface of the journals of a horizontal shaft, with ordinary oiling.

$V =$ the maximum permissible circumferential velocity of the journals of a shaft in feet per minute.

$C =$ A constant $= 28{,}000$, for cool bearings.

$C = 40{,}000$, for considerably heated bearings.

$C = 50{,}000$, for very hot bearings.

The maximum pressure per square inch, P, allowable with ordinary oiling on the horizontal bearing-surface of the journals of a shaft is, $P = C \div V$.

Example—A shaft having journals of 4 inches diameter, makes 100 revolutions per minute. What should be the maximum pressure on the bearings, consistent with freedom from liability to heat?

Then, the circumference of each journal of the shaft

= 4 × 3.1416 = 12.5664 inches ÷ 12 = 1.0472 feet; and 1.0472 feet × 100 = 104.72 feet per minute, the circumferential velocity of each journal of the shaft; and 28,000 ÷ 104.72 = 268 lbs. per square inch, the maximum pressure allowable on the horizontal bearing-surface of the journals. If the journals are 6 inches long, the horizontal area of bearing-surface = 4 × 6 = 24 square inches; and the bearing capacity of each journal is = 24 × 268 = 6,432 lbs.

Velocity of the Journals of a Shaft Suitable for a Given Pressure.—With ordinary oiling, the maximum permissible circumferential velocity in feet per minute of the journals of a horizontal shaft is V = C ÷ P, the notation being the same as in the preceding rule.

Example—A shaft with journals of 5 inches diameter has a pressure of 200 lbs. per square inch on the bearings. What should be the maximum circumferential velocity of the journals consistent with freedom from liability of the bearings to heat?

Then 28,000 ÷ 200 = 140 feet, the maximum permissible circumferential velocity of the journals per minute. The circumference of each journal of the shaft = 5 × 3.1416 = 15.708 inches ÷ 12 = 1.309 feet, and this shaft may make 140 ÷ 1.309 = 107 revolutions per minute.

Length of the Journals and Bearings of Shafts.—The length of the journals of shafts should be in proportion to the weight carried, and to the nature of the bearing-metal. Bearings of white metal, being softer, should be longer than those of bronze, in order to distribute the pressure over a greater surface. The following are good general proportions, in which D = the diameter of the journal in inches.

TABLE 22.—LENGTH OF THE JOURNALS AND BEARINGS OF SHAFTS FOR VARIOUS LOADS AT A SPEED NOT EXCEEDING 160 REVOLUTIONS PER MINUTE.

Description of Bearing-Metal	Length of Journal and Bearings in terms of the Diameter				
	For very light Loads	For light Loads	For ordinary or medium Loads	For heavy Loads	For very heavy Loads
Bronze and gun-metal....	D×1.00	D×1.25	D×1.50	D×1.75	D×2.0
Brass	D×1.25	D×1.50	D×2.00	—	—
Antifriction white-metal...	D×2.00	D×2.50	D×3.00	D×3.50	D×4.0
Cast-iron	D×4.00	D×4.25	D×4.50	—	—

The length of the journals and bearings of the spindles or shafts of emery wheels, fans, and blowers may be $= D \times 5\frac{1}{2} - 7$.

The length of the journals and bearings of the shafts of generators of electricity and other high-speed machines, may generally be as given in the following table:

TABLE 23.—LENGTH OF THE JOURNALS AND BEARINGS OF HIGH-SPEED MACHINES.

Revolutions of the Shaft in Feet per Minute.							
200	250	300	400	500	600	800	1000
D x 2.50	D x 2.75	D x 3.0	D x 3.5	D x 4.0	D x 4.5	D x 5.0	D x 6.0

Bearing Capacity of Journals.—The bearing-surface multiplied by the unit of pressure suitable for the particular bearing gives the bearing capacity of the journal. For instance, the projected area, or horizontal bearing-surface, of a journal of 4 inches diameter, and 6 inches long, is $4 \times 6 = 24$ square inches; and if a pressure of 300 lbs. per square inch be assumed, the bearing capacity of the journal is $24 \times 300 = 7{,}200$ lbs.

The length, L, in inches of the journal of a shaft is found as follows:

$$L = \frac{\text{Bearing capacity of the journal in lbs.}}{\left(\text{Pressure on the bearing in pounds per square inch}\right) \times \left(\text{width of the arc of contact of the bearing in inches.}\right)}$$

The width, W, in inches of the arc of contact of the bearing is:

$$W = \frac{\text{Bearing capacity of the journal in lbs.}}{\left(\text{Pressure on the bearing in pounds per square inch}\right) \times \left(\text{length of the bearing in inches.}\right)}$$

The pressure, P, in lbs. per square inch on the projected area of the shaft is:

$$P = \frac{\text{Bearing capacity of the journal in lbs.}}{\left(\text{Length of the bearing in inches}\right) \times \left(\text{width of the arc of contact of the bearing in inches.}\right)}$$

The usual pressure on various bearings of journals is given in Table 24.

Average Pressure on Various Bearings.—The average lowest, highest, and mean pressures on bearings in practice are generally as given in the following table:

TABLE 24.—PRESSURE ON THE BEARINGS OF THE JOURNALS OF SHAFTS, CRANK-PINS, CROSSHEAD-PINS, AND OTHER BEARINGS, MEASURED PER SQUARE INCH OF THE HORIZONTAL DIAMETRAL SECTION OF THE JOURNALS.

Description	Pressure in Pounds per Square Inch of the Horizontal Sectional Area of the Bearing		
	Lowest.	Highest.	Average.
Bearings of the shafting of factories, with couplings, but without pulleys.............	12	40	26
Bearings of the shafting of factories, with couplings, and with an average number of pulleys	30	100	65
Bearings of second-motion shafts, and main-shafting of factories, with couplings, pulleys, and toothed-wheel gearing	50	240	145

Final Alignment. After the pulleys have all been firmly keyed in their several places, the couplings set and properly bolted, and all of the journal bearings have been properly babbitted and the caps of the boxes tightened solidly upon the liners or bottom castings, as the case may be, the next test should be one of final alignment, not only for parallel lines of shafting, but pulley alignment also, in order that the belts when put on may run properly.

In the first place the main line shaft with a length, say of 50 feet and a weight of several tons of pulleys, etc., should, if properly adjusted, revolve so easily that one-man power applied to the rim of the 60-inch main pulley would turn it. If it cannot be so turned, it is an indication of abnormal tightness somewhere along the line, and the bearings should be carefully gone over again and examined. Some of them may need thicker liners. In order to have the belts run true on the pulleys of parallel shafting, it is absolutely necessary that the shafts shall not only lie level in their bearings, but the different lines be exactly parallel with each other, or at least as near so as it is possible to get them. This work may be done with the transit and station rod as already described, and in addition to this a great deal may be accomplished by the following-described method. Referring to Fig. 99, Shaft E is assumed to be level and it is desired to bring shaft F in line, or parallel with E. Pulley G is fastened in its permanent location on shaft E, and pulley H is also placed as near as possible in its proper location on shaft F by means of sighting across the rims of the pulleys. A line L is then stretched between points P and S across the two shafts and within a short distance, say one-half inch of the rim of pulley G at A and B.

The next step is to bring the rim of pulley H also within a half inch of the line at points C and D, which is done by adjustment of the bearings. Should the face of pulley H not be as wide as that of pulley G, allowance must be made in the measurements for the difference in width. Fig. 100 shows the two lines of shafting in perfect alignment, the center lines of both pulleys being at equal distance from the line L.

Having completed all adjustments by means of the line and measurements as above described a final transit test should be made on all the shafts, or in the absence of the transit, the test may be made with the straight-edge, and spirit level to ascertain whether or not the shafting is level.

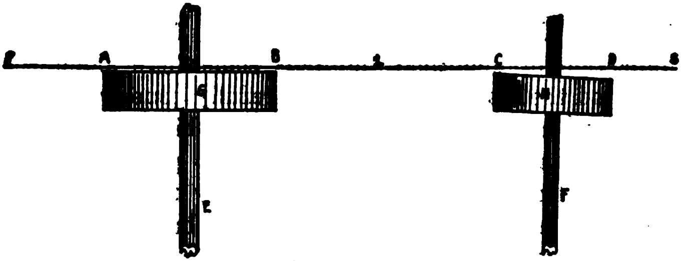


FIG. 99.

To test the alignment of the shaft with the permanent station stones, or targets, place the transit on the station stones and pick up the shaft-line, or take up that line from any targets that may be available for the purpose. The leveling-rod head may be either replaced on its rod, or put upon a piece of plain square rod. Place the free end of the rod against the shaft, hold it horizontal, and set the sliding head to a sight from the instrument cross hairs, then carry the rod to the next bearing, and if the cross hairs cut the same point on the rod, that section of the shaft may be considered all right. Proceed in the same manner to each bearing. Where a section of larger diameter is encountered, the rod must be shortened by moving the head towards the shaft a distance equal to the difference in diameters, and vice versa for a section of shafting of smaller diameter. Any variation found must be corrected by moving the bearing at which the variation is found, an amount sufficient to bring the shaft in line.

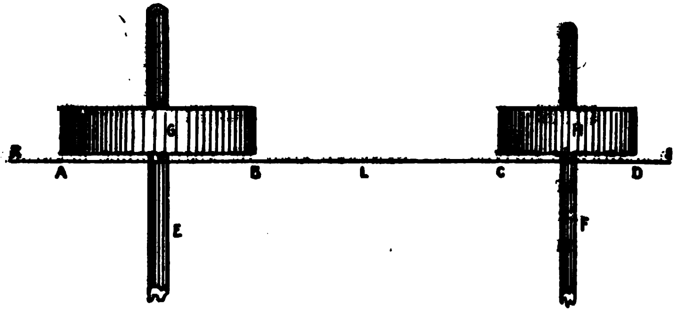
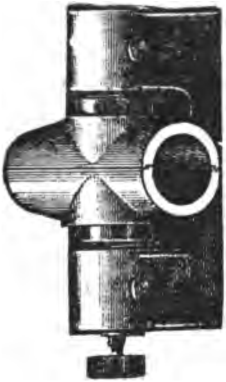


FIG. 100.

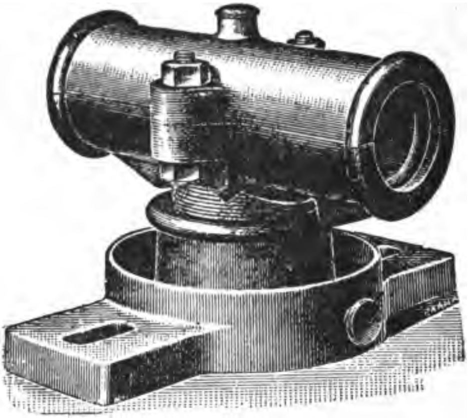
Fig. 100a shows several views of adjustable bearings for shaft journals, by the use of which the leveling of a line of shafting is made a comparatively easy task. The bearings shown are also capable of adjustment in a horizontal direction.

FRICTION OF BEARINGS.

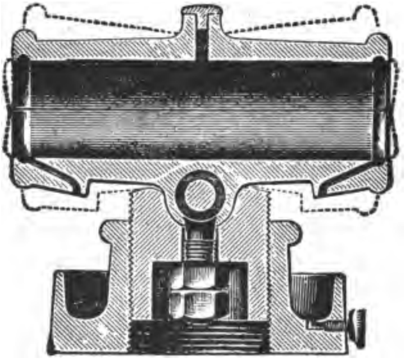
Thurston's Experiments. Prof. R. H. Thurston carried out a number of experiments to determine the effect of changes, not only in velocity, but also in pressure and temperature, upon the frictional resistance of lubri-



BAGSHAW'S SWIVEL ADJUSTABLE WALL-BRACKET BEARING.



GENERAL VIEW OF ADJUSTABLE BEARING.



VERTICAL SECTION OF ADJUSTABLE BEARING.

FIG. 100a.

cated bearings. His conclusions are, that the coefficient at first decreases, but after a certain point increases with the velocity; the point of change varying with the pressure and temperature.

Prof. Jenkins Experiments. A number of experiments were carried out by Prof. Jenkin, in connection with the difference between static friction, or the friction of rest, and dynamic friction, or the friction of motion. These experiments were made at extremely low velocities, and showed that in certain cases the coefficient of friction decreases gradually as the velocity increases, between the speeds of .012 and .6 foot per minute, thus indicating the probability of a continuous, rather than a sudden change in the value of the coefficient of friction between the conditions of rest and motion.

Beauchamp Tower's Experiments. In experimenting on the friction of lubricated bearings, and on the value of different lubricants, one of the difficulties which is first met with is the want of a method of applying the lubricant, which can be relied upon as sufficiently uniform in its action. All the common methods of lubrication are so irregular in their action that the friction of a bearing often varies considerably. This variation, though small enough to be of no practical importance, and to pass unnoticed, in the working of an ordinary machine, would be large enough utterly to destroy the value of a set of experiments, say, on the relative values of various lubricants; for it would be impossible to know whether an observed variation was due to a difference in the quality of the oil, or in its rate of application. The first problem, therefore, which presented itself, in the experiments, was to devise a method of lubrication such as would be perfectly uniform in its action, and would form an easily reproducible standard with which to compare other methods. These conditions were best fulfilled by making the bearing run immersed in a bath of oil. By this method the bearing is always supplied with as much oil as it can possibly take.

Proportions of Pulleys.—For the proportions of pulleys the following formulæ are presented by Unwin and Reuleaux: Let n = number of arms, D = diameter of pulley, S = thickness of belt, t = thickness of rim at edge, T = thickness in middle, B = width of rim; B^1 = width of belt, h = breadth of arm at hub, h^1 = breadth of arm at rim, e = thickness of arm at hub, e^1 = thickness of arm at rim, c = amount of crowning in inches; then,

	Unwin	Reuleaux
$B =$	$9/8\ (B^1 + 0.4)$,	$9/8\ B^1$ to $5/4\ B^1$
$t =$	$0.7\ S + .005\ D$,	(thickness of rim) $1/5\,h$ to $1/4\,h$
$T =$	$2t + C$,	
$h =$	for single belts $= .6337\sqrt[3]{\frac{BD}{n}}$ for double belts $= .798\sqrt[3]{\frac{BD}{n}}\ \frac{1/4''}{4} + \frac{B}{4} + \frac{D}{20\,n}$	
$h^1 =$	$2/3\,h$	$0.8\,h$
$e =$	$0.4\,h$	$0.5\,h$
$e^1 =$	$0.4\,h$	$0.5\,h$
$n =$ number of arms for a single set	$3 + \frac{BD}{150}$	$\frac{1}{2}(5 \times \frac{D}{2\,B}$

$$L = \begin{array}{l}\text{length}\\ \text{of hub}\end{array} \left\{\begin{array}{l}\text{not less than } 2.5\,S \\ \\ \text{is often } 2/3\,B\end{array}\right\} \begin{array}{l} B \text{ for single arm pulleys}\\ \\ 2\,B \text{ for double arm pulleys}\end{array}$$

M = thickness of metal in hub, h to $\frac{3}{4}\,h$
c = crowning of pulley $1/24\,B$

According to Unwin, the number of arms is arbitrary, and may be altered, if necessary. Reuleaux says, that pulleys with two or three sets of arms may be considered as two or three separate pulleys combined in one, except that the proportions of the arms should be 0.8 or 0.7 times that of single-arm pulleys.

Example—Dimensions of pulley as follows: Diameter 60″, face 16″, for double belt ½″ thick;

Solution by	n	h	h^1	e	e^1	t	T	L	M	c
Unwin . . .	9	3.79	2.53	1.52	1.01	.65	1.97	10.7	3.8	.67
Solution by Reuleaux. .	4	5.0	4.0	2.5	2.0		1.25	16	5	

The following solution for breadth of arm is proposed by Prof. William Kent: Assume a belt pull of 45 lbs. per inch of width of a single belt, that the whole strain is taken in equal proportions on one-half of the arms, and that the arm is a beam loaded at one end, and fixed at the other. The formula for a beam of elliptical section is $fP = .0982\,\dfrac{Rbd^2}{l}$, in which P = the load, R = the modulus of rupture of the cast iron, b = breadth, d = depth, and l = length of the beam, and f = factor of safety. Assume a modulus of rupture of 36,000 lbs., a factor of safety of 10, and an additional allowance for safety in taking $l = \frac{1}{2}$ the diameter of the pulley instead of $\frac{1}{2}D$ less the radius of the hub. Take $d = h$ the breadth of the arm at the hub, $b = e = 0.4\,h$, the thickness. We then have $fP = 10 \times \frac{45\,B}{n \div 2} = 900\,\frac{B}{n}$

$$= \frac{3535 \times 0.4\,h^2}{\frac{1}{2}D} \quad \text{whence} \quad h = \sqrt[3]{\frac{900\,BD}{3535\,n}} = \sqrt[3]{\frac{BD}{n}},$$

which is practically the same as the value reached by Unwin from a different set of assumptions.

Convexity of Pulleys.—On the subject of the convexity of pulleys, authorities differ. Morin gives a rise equal to 1/10 of the face; Molesworth, 1/24; others from ⅛ to 1/9. Scott A. Smith says, the crown should not be over ⅛-inch for a 24-inch face. Pulleys for shifting belts should be "straight," that is, without crowning.

Cone or Step Pulleys.—To find the diameter for the several steps of a pair of cone pulleys, proceed as follows: First, for crossed belts, let D and d be the diameters of two pulleys connected by a crossed belt, $L =$ the distance between their centers, and $B =$ the angle that either half of the belt makes with a line joining the centers of the pulleys; then total length of belt

$$= (D + d)\,\frac{n}{2} + (D + d)\,\frac{n\,B}{180} + 2\,L \cos B,$$

$$\beta = \text{angle whose sine is } \frac{D + d}{2\,L};$$

$$L \cos \beta = \sqrt{L^2 - {\frac{D + d}{2}}^{2}} \div L.$$

The length of the belt is constant when $D + d$ is constant; that is, in a pair of step pulleys the belt tension will be uniform when the sum of the diameters of each opposite pair of steps is constant. Crossed belts are seldom used for cone-pulleys, on account of the friction between the rubbing parts of the belt. To design a pair of tapering speed-cones so that the belt may fit equally tight in all positions, first, when the belt is crossed, use a pair of equal and similar cones tapering opposite ways; second, for open or uncrossed belts, use a pair of equal and similar cones tapering opposite ways and bulging in the middle, according to the following formula: Let $L =$ the distance between the axes of the conoids, $R =$ the radius of the larger end of each, $r =$ the radius of the smaller end; then the radius in the middle, r^1, is found as follows:

$$r^1 = \frac{R + r}{2} + \frac{(R - r)^2}{6.28\ L} \text{ (Rankine).}$$

If $D^1 =$ the diameter of equal steps of a pair of cone-pulleys, D and $d =$ the diameters of unequal opposite steps, and $L =$ the distance between the axes, then $D^1 = \dfrac{D + d}{2} + \dfrac{(D - d)^2}{12.566\ L}$. If a series of differences of radii of the steps $R - r$ be assumed, then for each pair of steps $\dfrac{R + r}{2} = r^1 - \dfrac{(R - r)^2}{6.28\ L}$, and the radii of each may be computed from their half sum, and half difference, as follows:

$$R = \frac{R+r}{2} + \frac{R-r}{2}; \quad r = \frac{R+r}{2} - \frac{R-r}{2}.$$

While it is good practice to run shafting at a high rate of speed, care must be exercised in order not to make it impossible to install the proper pulleys for driving the machines, at the correct speed. The speed of the shafting bears a direct relation to the size of pulleys required to obtain a certain increase or decrease in speed, and cases have been known in which the size of pulley necessary was so large that it could not be installed. This condition makes it imperative to obtain the correct speed by means of several reductions, and is not only a waste of power, but calls for extra belting, apparatus, lubricant and care.

Pulley Design. If a flexible band be wrapped completely about a pulley, and a heavy stress be put upon each end of the band, the rim of the pulley will tend to collapse just like a boiler tube with steam pressure on the outside of it. A compressive stress is induced which is very nearly evenly distributed over the cross-section of the rim, except at points where the arms are connected thereto. At these points the arms, acting like rigid posts, take this compressive stress. Now, a pulley never has a belt wrapped completely round it, the fraction of the circumference embraced by the belt being usually about ½, and seldom, even with a tightener pulley, reaching ¾. Assuming the wrap to be ½ the circumference, and that all the side pull of the belt comes on the rim, none being transmitted through the arms to the hub, we then have one-half of the rim pressed hard against the other half by a force equal to the resultant of the belt tensions, which, in this case, would be the sum of them. Dividing the pulley by a plane through its center and perpendicular to the belt, the cross-section of the rim cut by this plane has to take this compressive stress.

This analysis is satisfactory from an ideal standpoint only, for the intensity of stress due to the direct pull of the belt, with the usual practical proportions of rim, would be very small. Moreover, the element of speed has not been considered.

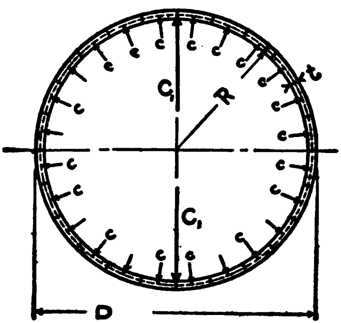


FIG. 101.

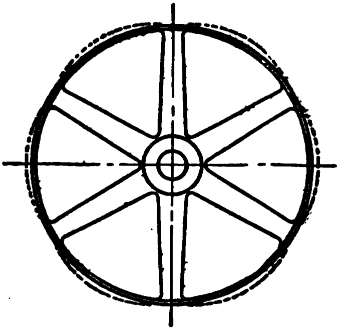


FIG. 101a.

When the pulley is under speed, a set of conditions which complicates matters is introduced. The centrifugal force due to the weight of the rim and arms is no longer negligible, but has an important influence upon the design and material used. This centrifugal force acts against the effect of the belt wrap, tending to reduce the compressive stress, or, overcoming the latter entirely, sets up a tensional stress both in the rim and in the arms. It also tends to distort the rim from a true circle by bowing out the rim between the arms, thus producing a bending moment in the rim, maximum at the points where the rim joins each arm.

In order to separate this action from other forces, it may be assumed that the rim is entirely free from the arms and hub, and is revolving about an imaginary center. Every particle now tends to fly radially outward from the center, owing to the influence of centrifugal force. This condition is shown in Fig. 101, where C, represents the force tending to separate the two halves of the rim, and c represents the radial force for each unit-length of rim. The case is similar to that of a boiler or a thin pipe subjected to a uniform internal pressure, which if carried to rupture, would split the shell along a longitudinal seam.

The tensile stress thus induced per square inch can be found by simple mechanics to be:

$$p = \frac{12wv^2}{g};$$

or, since $w = 0\,26$ pound, and $g = 32.2$ feet per second,

$$p = 0.097\,v^2 \quad \left(\text{say} \frac{v^2}{10}\right);$$

and, if p be taken equal to 1,000 pounds per square inch, which is as high as it is safe to work cast iron in this place,

$$v = 100 \text{ feet per second.}$$

This shows the curious fact that the intensity of stress in the rim is directly proportional to the square of the linear velocity, and wholly independent of the area of cross-section. It is also to be noted that 100 feet per second is about the limit of speed for cast-iron pulleys to be safe against bursting.

If we wish to consider theoretically the rim together with the arms as actually connected to it, we get a much more complicated relation. This condition is shown in Fig. 101a, where the rim, expanding more than the arms, bulges out between them. This makes the rim act something like a continuous beam uniformly loaded; but even then the resulting stress is not clearly defined on account of the variable stretch in the arms. Investigation on this basis is not needed further than to note that it is theoretically better, in the case of a split pulley, to make the joint close to the arms, rather than in the middle of a span.

Pulley Arms. The centrifugal force developed by the rim and arms tends to pull the arms from the hub. On the belt side, this is balanced to some extent by the belt wrap, which tends to compress the arm and relieve the tension. On the side away from the belt, the centrifugal action has full play, but the arm is usually of such cross-section that the intensity of this stress is very low. It may safely be neglected.

Pulley Face. To prevent the belt from running off, a crown, or rounding surface is given the rim. A tapered face, which is more easily produced in the ordinary shop, may be used instead. This taper should be as small as possible, consistent with the belt staying on the pulley. A taper of ½ inch per foot each way from the center is not too much for faces 4 inches wide and less, while above this width ¼ inch per foot is sufficient. As little as ⅛ inch total crown has been found to be sufficient on a 24 inch face, but this is probably too small for general service. Instead of being crowned, a pulley may be flanged at the edges, but flanged pulley rims chafe and wear the edges of the belt. The inside of the rim of a cast iron pulley should have a taper of ½ inch per foot to permit easy withdrawal from the foundry mould. This is known as "draft." If the pattern be of metal, or if the pulley be machine-moulded, the greater truth of the casting does not require that the inside of the rim be turned, as the pulley at low speeds will be sufficiently balanced to run smoothly. For roughly moulded pulleys, and for use at high speeds however, it is necessary that the rim be trued up on the inside in order to give the pulley a running balance. Fig. 102 shows a plain rim, a, also one stiffened by a rib, b. Where heavy arms are used this rib is essential, so that there will not be too sudden change of section at the junction of arm and rim, and consequent cracks, or spongy metal.

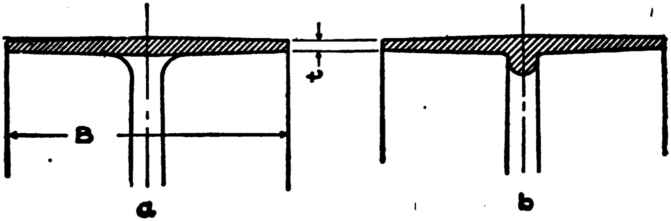


FIG. 102.

Belts and Belt drives. The use of belting in manufacturing plants, whether of rubber, leather, canvas or rope, brings many problems into the work of the millwright that require much judgment, skill and experience in order to produce satisfactory results at a minimum cost and keep the production factor constant. Any special local conditions in every plant have to be met by a special method of procedure. The speed of belts should be between 4,000 and 5,000 feet if possible, as better results may be obtained with smaller belts under these conditions, and this factor of velocity should enter into whatever belt formula is used in determining the horsepower, or the width of the belt.

The atmospheric conditions are as variable as the power factor, and the life and efficiency of the driving medium are almost entirely governed by these conditions. In cotton mill work the air is dry and linty, and even with humidifiers working, the belts will occasionally need a liberal amount of neatsfoot oil or other good dressing. Belts for this work are best when made of leather, and what is known as a "soft tan" gives excellent results as regards pliability.

Machine shops are generally dusty and dry as regards atmospheric conditions, and wood working shops polishing and grinding rooms are invariably so. Even under these conditions if belts are kept soft and pliable, and well laced or cemented they will last almost indefinitely.

In designing belt drives there are several very important features which the millwright should keep constantly before him, and correct data upon these points should be obtained before any figuring is done. For instance he should know:

(a) What is the distance between centers.
(b) The kind of pulleys—iron, wood, paper, etc.
(c) The kind of belt—leather, rubber, cotton, etc.
(d) Is the belt to be single, double or treble?
(e) What velocity can be given the belt?
(f) What horse power is to be transmitted?
(g) Will the resistance to turning be constant?

The tension per inch of belt width that may be allowed is also an important factor. Some designers have adopted 40 lbs. pull per inch of width as a standard for all kinds of belts, leather, rubber and cotton. Others allow 60 lbs., while others still allow 88 lbs., but only for certain kinds of belting. A belt transmits motion entirely through frictional contact with the surface of the pulley.

It is best to have the face of the pulley from ½ inch to 1½ inch wider than the belt, depending upon width of face. Wide belts are less effective per unit of sectional area than narrow belts, and long belts are more effective than short ones. The proportion between the diameters of two pulleys working together should not exceed 6 to 1.

The usual convexity of pulley face, is ⅛ to ¼ inch in the width, up to 12 inches wide, and for larger sizes ⅛ to ¼ in. per foot of width.

Speed of Belts.—Belts have been employed running over 5,800 feet per minute, but nothing is gained by running belts at a speed much in excess of 4,000 feet per minute. About 3,500 feet per minute for main belts is good practice, and for lathe belts from 1,500 to 2,000 feet per minute. Table 25 gives the coefficients of friction of different kinds of belts.

Table 25

Coefficient of Friction.

Leather belt on iron pulley, dry	0.42
Leather belt on iron pulley, wet	0.36
Leather belt on iron pulley, greasy	0.23
Leather belt on leather covered pulley	0.45
Cotton belt on iron pulley, aver.	0.30
Rubber belt on iron pulley, aver.	0.45

At high velocities the centrifugal tension will decrease the pressure between belt and pulley, besides putting additional tension in the belt.

Rankine's formula for centrifugal tension is

$$C = \frac{12 \times W \times V^2}{g}.$$

in which W = Weight of cu. in. of leather,
C = Centrifugal tension,
V = Velocity in feet per second,
g = Acceleration of gravity = 32.2,
let $W = .035$,

$$\text{then } C = \frac{12 \times .035 \times V^2}{32.2} = \frac{.42 \times V^2}{32.2} = .013V^2.$$

On high speed belts running at, say over 3,000 feet per minute, the effect of centrifugal tension becomes an important factor. For instance, suppose we had a 6 in. belt ¼ in. thick, with cemented joints; Max. safe load to be 400 lbs. per sq. in. Then $6 \times .25 = 1.5$ sq. in., which $\times 400 = 600$ lbs. If this belt were to be speeded up to 176 feet per second, then C will equal $.013 \times 176^2 \times 1.5 = 604$ lbs., and therefore the centrifugal force would equal the safe tension and the belt could not transmit any power. Thus it is necessary to figure on centrifugal force for all speeds of 3,000 feet per minute or more.

Take another example; a 20-inch belt ¼ inch in thickness is to run at a speed of 2,670 feet per minute, and the pull on the belt is 1,235 lbs., equal to $1{,}235 \div 20 = 61.8$ lbs. pull per inch width. To this must be added the centrifugal tension. The belt being 20 inches wide by ¼ in. thick, there are $20 \times .25 = 5$ sq. inches of leather in section, then $C = .013 \times 44.5^2 \times$

5 = 128.7 lbs. This will increase the total tension on the belt to 1,235 + 128.7 = 1,363.7 lbs., and the pull per inch width of belt will be 1,363.7 ÷ 20 = 68.1 lbs.

Strength of, Working Tension in, and Horse Power Transmitted by, Belts.—The ultimate strength of ordinary bark tanned single leather belting varies from 3,000 to 5,000 lbs. per square inch of cross-section. It is, however, sometimes considered more convenient to state the tenacity of a belt in lbs. per inch of width. The thickness of single belting varies from 3/16 to 5/16 inch, and from ⅜ to ⅝ inch for double belting. Taking the mean thicknesses, we may say that the breaking stresses are from 750 to 1,250 lbs. per inch of width for single belts, and 1,500 to 2,500 lbs. per inch of width for double belts.

It is found, however, that the strength of the joints is sometimes only about one-third of the strength of the solid leather. Hence, the *final* strength of an endless belt should not be reckoned at more than *one-third* of the above values. Further, the *safe working tension* should never exceed *one-fifth* of this final strength (of the joint), in order to provide for deterioration and sudden changes of load. From this point of view, we thus arrive at approximate values for the safe working tensions by taking one-fifteenth (*i.e.*, one-third for joint and one-fifth for factor of safety) of the breaking stress *per inch of width*, viz.:

For single belting,	50 to 80 lbs.
For double belting,	100 to 160 lbs.

In *practice,* however, the rule is not to put on a greater tension than 50 lbs. for single, and 80 lbs. for double belts per inch of width, since it is found that under these mild conditions, leather belts run for many years with a minimum of lubrication on the bearings, and with far less chance of heating the journals or stressing the shafts.

Power of Single Leather Belts.—To roughly calculate the power of single leather belts, the following formula may be used: Let H. P. = actual horse-power; W = width of belt; F = driving force; T = working tension from 70 to 150 lbs. per inch of width; V = velocity of belt in feet per minute. Then

$$F = \frac{33{,}000 \times H.P.}{V}, \text{ or } F = \frac{W \times T}{2}. \quad H.P. = \frac{V \times F}{33{,}000}.$$

$$W = \frac{33{,}000 \times H.P.}{\frac{T}{2} \times V}.$$

These rules give good practical results where there is no great inequality in the diameter of the pulleys.

Table 26 will be found very useful in estimating the horse power that different sized belts will transmit. For instance a single oak tanned leather belt running at a speed of 700 feet per minute will transmit 1.06 horse power per inch in width, while a heavy double belt of the same material and at the same speed will transmit 1.91 H.P.

Which Side of the Leather Should Face the Pulley.—The grain or hair side of the leather is naturally much smoother than the flesh side, and millwrights differ in opinion regarding which of these sides should run in contact with the pulley. Whichever side of the belt is in contact with one pulley, the same side should

TABLE 26

HORSE-POWER THAT DIFFERENT LEATHER BELTS WILL TRANSMIT PER INCH IN WIDTH AT VARIOUS SPEEDS.

Velocity of Belt per Minute.	KIND OF BELTS.								
	Best Oak-tanned Belts.			Best Link or Chain Belts.					
	Single belts.	Light double belts.	Heavy double belts.	⅜″	½″	⅝″	¾″	⅞″	1″
Feet.	HORSE-POWER THEY WILL TRANSMIT.								
100	·15	·21	·27	·13	·15	·17	·20	·24	·27
200	·30	·42	·55	·25	·29	·35	·40	·47	·55
300	·45	·64	·82	·38	·44	·52	·60	·71	·82
400	·61	·85	1·09	·51	·58	·69	·80	·95	1·09
500	·76	1·06	1·36	·64	·73	·86	1·00	1·18	1·36
600	·91	1·27	1·64	·76	·87	1·04	1·20	1·42	1·64
700	1·06	1·49	1·91	·89	1·02	1·21	1·40	1·65	1·91
800	1·21	1·70	2·18	·92	1·16	1·38	1·60	1·89	2·18
900	1·36	1·91	2·45	1·05	1·31	1·55	1·80	2·13	2·45
1000	1·51	2·12	2·73	1·27	1·45	1·73	2·00	2·36	2·73
1100	1·67	2·33	3·00	1·40	1·60	1·90	2·20	2·60	3·00
1200	1·82	2·55	3·27	1·53	1·75	2·07	2·40	2·84	3·27
1300	1·97	2·76	3·55	1·65	1·89	2·25	2·60	8·07	3·55
1400	2·12	2·97	3·82	1·78	2·04	2·42	2·80	3·31	3·82
1500	2·27	3·18	4·09	1·91	2·18	2·59	3·00	3·55	4·09
1600	2·42	3·39	4·36	2·04	2·33	2·76	3·20	3·78	4·36
1700	2·58	3·61	4·64	2·16	2·47	2·94	3·40	4·02	4·64
1800	2·73	3·82	4·91	2·29	2·62	3·11	3·60	4·25	4·91
1900	2·88	4·03	5·18	2·42	2·76	3·28	3·80	4·49	5·18
2000	3·03	4·24	5·45	2·55	2·91	3·45	4·00	4·73	5·45
2100	3·18	4·45	5·73	2·67	3·05	3·63	4·20	4·96	5·73
2200	3·33	4·67	6·00	2·80	3·20	3·80	4·40	5·20	6·00
2300	3·49	4·88	6·27	2·93	3·35	3·97	4·60	5·44	6·27
2400	3·64	5·09	6·55	3·05	3·49	4·15	4·80	5·67	6·55
2500	3·79	5·30	6·82	3·18	3·64	4·32	5·00	5·91	6·82
2600	3·94	5·52	7·09	3·24	3·78	4·49	5·20	6·15	7·09
2700	4·09	5·73	7·36	3·28	3·85	4·66	5·40	6·38	7·36
2800	4·24	5·94	7·64	3·31	3·86	4·73	5·60	6·62	7·64
2900	4·39	6·15	7·91	3·32	3·87	4·78	5·80	6·85	7·91
3000	4·50	6·36	8·18	3·31	3·86	4·75	5·97	7·09	8·18
3100	4·60	6·58	8·45	3·30	3·85	4·73	5·96	7·33	8·45
3200	4·69	6·79	8·70	3·28	3·82	4·71	5·94	7·37	8·73
3300	4·77	7·00	8·86	3·24	3·77	4·70	5·92	7·35	8·88
3400	4·84	7·21	8·96	3·19	3·71	4·64	5·87	7·32	8·86
3500	4·90	7·31	9·06	3·13	3·61	4·50	5·78	7·26	8·80
3600	4·95	7·40	9·16	3·05	3·50	4·37	5·67	7·16	8·73
3700	4·99	7·48	9·24	2·96	3·39	4·26	5·55	7·01	8·58
3800	5·03	7·54	9·29	2·84	3·28	4·15	5·41	6·87	8·41
3900	5·03	7·60	9·34	2·72	3·13	4·02	5·20	6·70	8·27
4000	5·08	7·64	9·37	2·58	2·95	3·84	5·01	6·48	8·04

be in contact with all the pulleys over which it passes, so as not to bend it alternately in different directions. When the flesh side is next to the rim of the pulley, it should receive one coating of currier's dubbing, and three coatings of boiled linseed oil every year, in the case of a belt having to endure continuous hard work. This has the effect of rendering it about as smooth as the hair side, and its efficiency as a whole is said to be thereby increased. When the hair side is in contact with the pulley, it is usual to give that side an occasional coating of castor oil in order to render it more flexible, and, consequently, more durable than it would be in its natural state. Rosin or cobbler's wax should *never* be employed, as they gather dirt, and form lumps on the pulley and belt.

Double and Treble Belting.—Sometimes the breadth of a single belt necessary to transmit a given power would be inconveniently great, and hence double or treble belts are used. These are made by cementing, and then riveting, or sewing two, or three thicknesses of belting together. Owing to the greater rigidity of such belts they do not work so satisfactorily as single ones, and are not well adapted for running at high speeds, or over pulleys having a diameter of less than 3 or 4 feet, or in cases where the pulleys are close together.

Woven Belting.—Although leather belting is still very largely employed, it has, more especially during recent years, been very largely displaced by woven beltings, which are strong and very pliable—two of the most important features in any flexible transmitter of power—while they have the further advantage of not being affected by changes of temperature, water, steam, fumes of chemicals, etc.

Belt Tension. The tension in a belt is nearly uniform throughout its length so long as it is at rest, except in cases where the pulleys are a considerable distance apart when the weight of the belt will produce a slightly greater tension next to the pulleys than there is in the middle of the span. But when power is being transmitted through the belt the condition of stress in the belt is at once materially changed. As the belt is a flexible member, we can transmit only a pull to the other pulley, thereby turning it around, the push which is at the same time given to the other side of the belt merely acting to make the belt sag or become slack. Hence the immediate effect of starting motion in a belt is to change the condition of equal tension throughout its length, to that of unequal tension in the two sides. The driving side is tight, while the other is loose, the former having gained as much tension as the latter has lost, and the sum of the two being practically equal to the sum of the tensions in the two sides of the belt when at rest. This is not strictly true, as will be shown later; but it is sufficiently accurate to form a good basis for the practical design, at least of slow-speed belts.

This condition of tight and slack sides is made possible by the fact that the belt, in being wrapped around the pulleys under tension, has friction on their surfaces. Thus, we can pull hard on one side without slipping the belt around the pulleys, but could not do this if the pulleys were perfectly smooth or frictionless, for in that case the slightest pull on one side would slip the belt around the pulleys. In fact, it would be impossible to produce any pull by means of the driving pulley, for the pulley would merely slip around inside the belt.

The amount of pull we can apply to the belt is therefore limited by the tension at which the belt slips around the pulley. Moreover, since the force of friction between the belt and pulley is dependent upon the normal force with which the belt is pressed against the pulley, and the coefficient of friction between the two, it is evident that the tighter the belt is laced up, and the rougher the surfaces of the pulley and belt, the greater is the force that can be transmitted through the belt.

Again, when the belt, as it passes around the pulley, changes its straight-line direction to circular motion, each particle of the belt—like a body whirling at the end of a cord about a center of rotation—tends by centrifugal force to fly away from the surface of the pulley, thereby decreasing the normal pressure, and hence the friction. This centrifugal force also changes somewhat the tensions in the belt between the pulleys. As the centrifugal force increases in proportion to the square of the linear velocity, it is evident that the effect is greater at high speeds than at moderate or low speeds.

A further circumstance that affects the driving power of a belt is the stiffness of the leather or other material of which the belt is made. As it passes around the pulley, the belt is bent to conform to the circumference of the pulley, and is again straightened out as it leaves the pulley. Hence the theoretically perfect action is modified somewhat according to the sharpness of the bending and the thickness or flexibility of the belt; in other words, a small pulley carrying a thick belt would be the worst case for successful calculation on a theoretical basis.

Strength of Leather Belting. The breaking tensile strength of leather belting varies from 3,000 to 5,000 pounds per square inch. Joints are made by lacing, by metal fasteners, or by cementing. The strength of a laced joint may be about 7/10, of a metal-fastened joint, about ½, and of a cemented joint, about equal to the full strength of the belt cross-sectional area. The proper working strength of belting depends on the use to which the belt is put. A continuously running belt should have a low tension in order to have long life and a minimum loss of time for repairs. For double leather belting it has been shown that a working tension of 240 pounds per square inch of sectional area gives an annual cost—for repairs, maintenance, and renewals—of 14 per cent of first cost. At 400 pounds working tension, the annual expense becomes 37 per cent of first cost. These results apply to belts running continuously; larger values may be used where the full load comes on but a short time, as in the case of dynamos.

Good average values for working tensions of leather belts are:

Cemented joints,	400 pounds per square inch.
Laced joints,	300 pounds per square inch.
Metal joints,	250 pounds per square inch.

Speed of Belting. The most economical speed is somewhere between 4,000 and 5,000 feet per minute. Above these values the life of the belt is shortened; also "flapping," "chasing," and centrifugal force cause considerable loss of power. The limit of speed with cast-iron pulleys is fixed at the safe limit for bursting of the rim, which may be taken at one mile per minute.

Material of Belting. Oak-tanned leather, made from the part of the hide which covers the back of the ox, gives the best results for leather belting. The thickness of the leather varies from .18 to .25 inch. It weighs from .03 to .04 pound per cubic inch. The average thickness of double leather belts may be taken as .33 inch, although a variation in thickness from ¼ inch to 7/16 inch is not uncommon. Double leather belts may be ordered light. medium, or heavy.

In a single-thickness belt the grain or hair side should be next to the pulley, for the flesh side is the stronger and is therefore better able to resist the tensile stress due to bending set up where the belt makes and leaves contact with the pulley face. Double leather belts are made by cementing the flesh sides of two thicknesses of belt together, leaving the grain side exposed to surface wear.

Raw hide and semi-raw hide belts have a slightly higher co-efficient of friction than ordinary tanned belts. They are useful in damp places. The strength of these belts is about one and one-half times that of tanned leather.

Cotton, cotton-leather, rubber, and leather link belting are some of the forms on the market, each of which is especially adapted to certain uses.

General Arrangement of Belts and Pulleys. If possible machinery should be placed in such a manner that the direction of the belt motion shall be from the top of the driving pulley, because then the sag of the belt will increase the arc of contact of the belt with both the driving, and driven pulleys. When arranging the belts leading from the main line of shafting to the countershafts, those pulling in an opposite direction should be placed as near each other as practicable, while those pulling in the same direction should be separated from each other. By pursuing this method the friction on the journals may be greatly decreased, and it can often be accomplished by changing the relative positions of the pulleys on the counter shafts. The angle of the belt with the floor should not exceed 45 degrees. Shafts connected by belting should never be placed one directly over the other, if it is possible to avoid it, because in such cases the belt must be kept very tight to do the work. Another very important point to be observed in the arrangement of belts and pulleys is to so locate the shafts, that there will be sufficient distance between them to allow of a gentle sag in the belt while in motion. Where narrow belts are to be run over small pulleys, 15 feet is a good average distance between shafts. For larger belts the distance should be 20 to 25 feet, and for main belts the distance should be not less than 25 to 30 feet, allowing a sag of 4 or 5 inches.

Belt Fastenings. The best fastening for a belt is the cement splice. It is much better than lacing, belt hooks, riveting, or any other method of joining together the ends of a belt. Good glue makes a strong and durable cement for leather belts, but fish glue is less affected by moisture. Ordinary hot glue may be made impervious to water by adding a small quantity of bichromate of potash to the glue just before using.

Lacing Belts.—Joining the ends of a belt by lacing tends to weaken the belt by just the amount of material that is punched out for the holes that are required to receive the lacing. The best method to pursue in lacing is, to use as small lacing as possible, and pass it many times through small holes, rather than a wide lacing passed through large holes. As an instance of the weakening effect of punching holes of too large diameter in a belt for the purpose of lacing it, the following example may be cited. Taking a double belt 5 inches in width, although the width is immaterial as the loss of strength is calculated as a given per cent of total width. There are four holes in this piece of belt, each hole 3/8 inch in diameter. The aggregate width thus cut out of the belt is 4 × ⅜ inches = 12/8 = 1½ inches. Then 1.5 ÷ 5 = 0.30, or 30 per cent of the belt has been cut away—nearly one-third of the total strength.

Now if in place of a few large holes there are more smaller ones—one-fourth more, in fact. There are five holes, each 3/16 inch in diameter, making a total of 15/16 inches, or 0.9375 ÷ 5 = 18¾ per cent, leaving 81¼ per cent of the total belt strength against 70 per cent in the belt with large holes. A first-class double leather belt will tear in two under a strain of about 500 pounds to each lace hole, the strain being applied in the holes by means of lacings.

Thus it would require 2,500 pounds to tear the 5-inch belt with the small lace holes, while the belt with the large holes would only stand 2,000 poundts.

Another method of computing the working strength retained in the belt after punching the holes is, assuming the tensile strength of the leather to be 3,000 lbs. per sq. in., the 5-inch belt being ⅜ in. thick will have a cross sectional area of 5 × 0.375 = 1.875 sq. in. After deducting 30 per cent of this for the holes, there still remains a belt section of 1.125 sq in., having a breaking strength of 1.125 × 3,000 = 3,375 lbs. But as the large lacing and lace holes of the belt broke it under a strain of 2,000 lbs., it is plainly evident the strain is not evenly divided. With the other method of lacing, viz: small holes, the belt retains 81¼ per cent of its original cross section = 1.525 sq. in. after punching the five small holes. The breaking strain in this case = 3,000 × 1.525 = 4,575 lbs., using a factor of safety of 5 the working strain = 4,575 ÷ 5 = 915 lbs. per sq. in., or 1.525 × 915 = 1,395.375 lbs. for the belt. There being 5 lace holes, the pull on each lace = 1,395.375 ÷ 5 =

279 lbs., and using a factor of safety of 2 for the lacing, the safe working strength of which = 500 ÷ 2 = 250 lbs., it will be seen that the stress is more evenly divided by the use of the small holes and a larger number of them.

FIG. 102a.

Fig. 103 shows a variety of belt joints. At (1) is shown a joint which is generally made in a belt factory where the necessary scarfing planes, and compressing gear are available. Cemented joints made in this way are nearly as strong as riveted joints. A good connector for new and strong leather belts is shown at

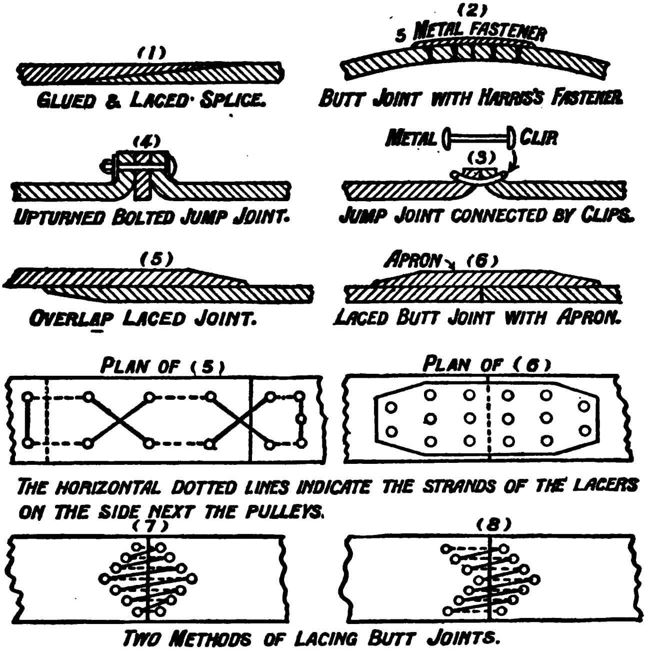

(1)
GLUED & LACED· SPLICE.
(2)
5 METAL FASTENER
BUTT JOINT WITH HARRIS'S FASTENER.
(4)
UPTURNED BOLTED JUMP JOINT.
METAL CLIP
(3)
JUMP JOINT CONNECTED BY CLIPS.
(5)
OVERLAP LACED JOINT.
APRON
(6)
LACED BUTT JOINT WITH APRON.
PLAN OF (5)
PLAN OF (6)
THE HORIZONTAL DOTTED LINES INDICATE THE STRANDS OF THE LACERS ON THE SIDE NEXT THE PULLEYS.
(7)
(8)
TWO METHODS OF LACING BUTT JOINTS.

FIG. 103.
VARIOUS METHODS OF JOINING LEATHER BELTS.

(2). The tough yellow metal fasteners shown at (3) serve well for belts running at high speeds over small pulleys.

The slits for these clips are made by a special form of cutting pliers and should be cut fully ¼ inch from the end of the belt, being so spaced that the crossheads of the fasteners come close to each other across the strap. In order to prevent the body of the metal clip from being bent up and down when passing over the pulleys, the fastener should be as short as possible.

At (4) is shown what is termed a "jump joint" upturned and bolted. The strength of this joint is

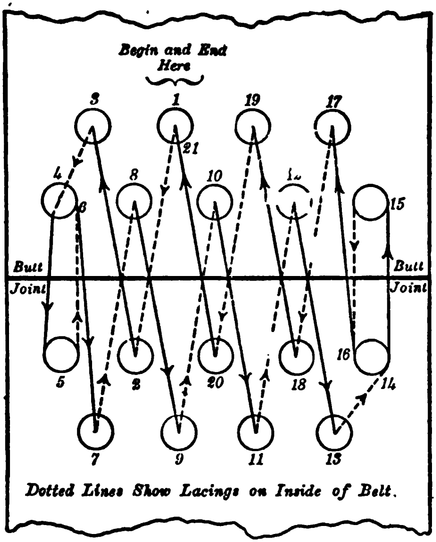


FIG. 104.
PLAN OF A LACED BUTT JOINT FOR 6-INCH BELT.

increased by inserting a leather strip or washer between the turned up ends as shown. An overlap lace joint in both section and plan is shown at (5). In making this joint, care should be taken to taper down and curve the ends to suit the smallest pulley over which the belt has to pass, otherwise the joint will be stiff, and consequently every time it travels on and off the pulley a sort of hinge action takes place, accompanied by a shock which not only shortens the life of the belt, but also communicates a jar to the shaft and machine being driven.

For heavy broad double belts where a scarfed cemented joint cannot be used, the form of joint shown in plan and section at (6) with leather apron makes a very good connection; (7) and (8) show the best methods of lacing simple butt joints. The holes are punched so as to form a diamond or pointed figure, whereby there are never more than two holes in line across the belt. Consequently it is possible to retain almost the entire strength of the belt without impairing its flexibility. Another plan of lacing a butt joint, which is very suitable for dynamo driving, and insures steady, smooth running, is illustrated in Fig. 104.

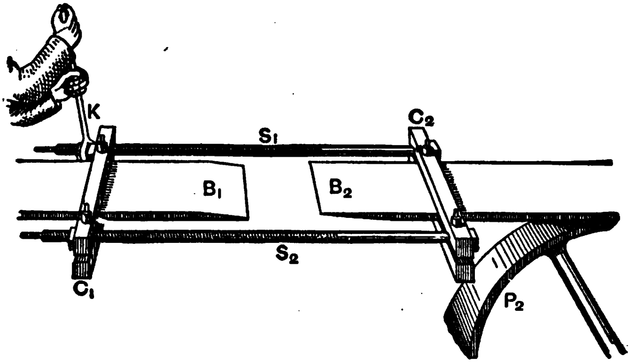


FIG. 105.
BELT STRETCHER.

This joint is made by a lace in one length by beginning at hole 1, and continuing through the several holes in the order 2, 3, 4, etc., as shown in Fig. 104, ending with hole 1, which is also marked 21.

In order to facilitate the making of any one or other of the above-mentioned joints in belts more than 6 inches wide, it is usual to employ a "drawing-up frame" of the form shown by Fig. 105. The two ends, B_1, B_2, of the belt, after being passed over the driving and driven pulleys, are brought towards each other and gripped by the clamps, C_1, C_2. The wrench, K, is then applied alternately to the nuts of the long screws, S_1, S_2, until the tapers of the splice are brought fairly over each other, or until the ends come together fairly and squarely in the case of a butt joint. The lacing can then be done without any other effort than that of merely passing the thong through the holes and fastening it.

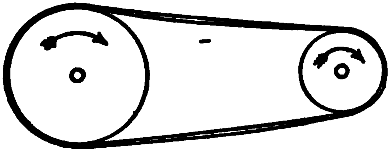


FIG. 106.
COMPOUND BELT DRIVING.

Compound Belting.—In order to avoid the internal straining action of thick double or treble belting, it has been found advisable to simply put two or three thin single belts on the top of each other, without any cementing or riveting together, as shown by Fig. 106. This gives perfect freedom of action to each belt to accommodate itself to the curves over which it passes without stressing its neighbours. The friction between the surfaces of these several belts is not found to have any observable objections, and it is said that 70 per cent more power can be transmitted by compounding two single belts in this manner.

By the addition of a third belt, still more power may be transmitted. A simple modification of this compound belt drive is illustrated by Fig. 107. The power is transmitted from one main driving pulley to two or more driven pulleys by separate belts, all moving in the same plane, and in a direct line with each other. This arrangement saves considerable room in a workshop or factory. It has been proved to be very convenient for driving several dynamos direct from one flywheel, instead of from independent drums.

Link Chain Belting.—This form of belting (see Fig. 108) permits of the escape of the air through the spaces between the links, and thus enables the leather to bear uniformly over the face of the pulley.

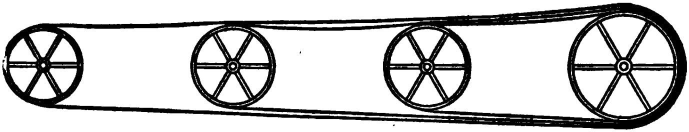


FIG. 107.
COMPOUND BELT DRIVE FOR THREE MACHINES.

It is composed of a series of short leather links bound together by steel pins and washers. These belts possess considerable flexibility, and can be made with a flexible central row of links to automatically suit the arch of any pulley. If desired, they may be specially curved to suit any camber. They further possess the advantage of being easily shortened and rejointed by simply bringing the ends together with a drawing-up frame, interlocking the links until a row of rivet holes are brought fair in a line, and then inserting the rivet. When one side of this belting is made thicker than the other it is well adapted for working on tapered cones and for quarter twist driving.

A Good Belt Dressing.—A good formula for a surface compound for belts is: Equal parts of red lead, black lead, French yellow and litharge. Mix with boiled linseed oil and add enough japan to make it dry quickly. A thin coating can be applied with a brush and should be allowed to dry before running the belt. There are several good preservative foods or dressings manufactured by reliable firms which may be applied to belts after cleaning and from which splendid results have been obtained, as shown by experiments, records and data kept before and after treatment.

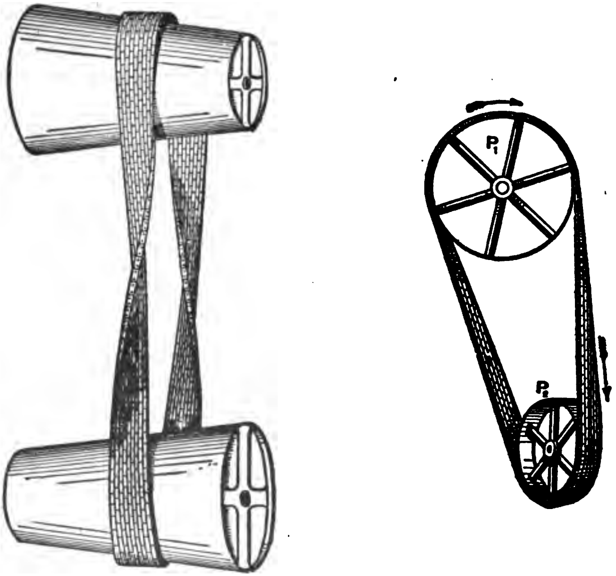


FIG. 108.
CHAIN BELTING—CHAIN BELTING WORKING QUARTER TWIST.

Belt Shifters.—A shifter having rollers should be used when the belt is a wide one. These rollers should press against the flat of the belt, not the sides, thus avoiding heating and damaging of the edge of the belt. Near each pulley on the lineshaft, where a belt drives a machine that is liable to stand idle for some time, a staple should be driven into the rafters or ceiling. A hook of 3/8-in. round iron should be hung from the staple; it should be made long enough to reach almost to the rim of the pulley, but a little to one side of it. When a belt is taken off its pulley it should be hooked up by the belt stick, the slack of the belt allowing it to be caught on the hook. When the belt sags the hook should swing so as to make the belt clear the side of the pulley and also the shaft, as shown in Fig. 109. This avoids the tieing of belts to beams, hangers, etc., or of leaving them hanging on the revolving shaft, causing the belt to be worn through or weakened. It is very little trouble to hook them up.

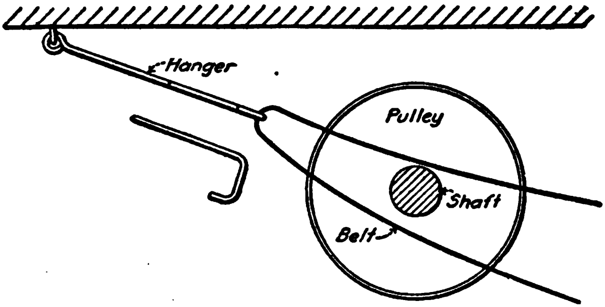


FIG. 109.

HANGING AN IDLE BELT TO CLEAR PULLEY AND LINE SHAFT.

Belt Guards.—All belt drives within reach of persons standing on the floor, or on adjacent platforms, such as drives to emery wheels, etc., should be carefully guarded by wire screens of not less than 1-in. mesh. These are light and can be easily removed when it is necessary to get at the belt.

General Suggestions.—1. Never overstrain a belt, as this produces unnecessary wear of belts and machinery and causes great loss of power by friction.

2. Do not throw on belts when pulleys are running at an extremely high rate of speed.

3. Do not run belts exceedingly tight, as the best service and greatest power are derived by their being just slack enough not to slip.

Rope Transmission.—The transmission of power is frequently accomplished by means of ropes instead of belts. In addition to other advantages of this method, ropes seldom give way without due warning by slackening or fraying in one or more of the strands, thus reducing the risk of accident, and stoppage of the mill or factory to a minimum. The working stress of the ropes being but a small fraction of their breaking strength, any signs of weakness in an individual rope would allow it to be removed, and the engine run with the remaining ropes until a convenient opportunity is offered for the replacement of the weak member. Further their comparative slackness between the pulleys facilitates their annulling any small irregularity in the motive power. These ropes are made of manilla-hemp, cotton, leather, and raw ox hide. Hemp ropes are preferred to cotton ropes for main drives with large pulleys, since they are cheaper, stronger, and last nearly as long as if spun with a greased core.

Cotton ropes, being softer and more pliable than manilla ropes, can be used with smaller pulleys without undue injury to the fibres. This is also considerably aided by the natural wax in the structure of the long staple variety which acts as a lubricant, and permits of greater freedom of motion between the several fibres. The life of a good cotton rope is usually about thirteen years if properly adjusted and well treated.

Sizes of Ropes and Pulleys.—The size and number of ropes, as well as the least diameter of pulley, for any given power, are points of importance and should be considered for each case.

The ropes commonly used for the transmission of power in factories or mills vary from 3 to 5 inches in circumference. No matter what the diameter of the pulley may be, ropes of 1¾ inches diameter should not be exceeded for main drives, and 1¼ inches diameter for secondary drives. The diameter of the smallest pulley should not be less than thirty times the diameter of the rope, as the larger the pulley the less will be the internal friction, and consequent injury to the rope from bending and unbending (see table 27).

Strength of Cotton and Hemp Ropes.—The ultimate strength of white untarred cotton ropes may be taken at 9,000 lbs. per square inch of the net sectional area, which is about 90 per cent of the area of the circumscribing circle. The normal working stress should not exceed 1/30 of the ultimate breaking stress, or say 300 lbs. per square inch, although ropes are frequently worked at even a less tension.

Table 27 gives useful data regarding these points. The ultimate strength of *new manilla* ropes is about 11,000 lbs. per square inch of the net sectional area, which in a three stranded hawser is only about 80 per cent of the arc of its circumscribing circle. The necessary lubrication, however, reduces the strength by 20 to 30 per cent., but the lubrication of the fibres is of much greater importance than the actual breaking stress. The greater freedom of movement amongst the fibres permits a heavier working stress to be carried, and ensures a much longer life. A properly lubricated manilla rope will outlast from two to four similar dry laid ropes working under the same conditions. There are many ways of lessening the internal friction, one of the best being that of coating the several yarns with a mixture of black lead and tallow prior to twisting the same into strands. Under favorable conditions, when thus treated, it is practically waterproof, and will last for about eight years.

TABLE 27.—COTTON DRIVING ROPES AND PULLEYS.

Dia-meters. Ins.	Areas. Ins.	Weight per Foot. Lbs.	Ordinary Working Tension including Centrifugal Force. Lbs.	Velocity of ropes, 4,700 ft. per minute (their most effective speed).			Rope Pulleys.	
				Stress due to Centrifufal Force Lbs.	Effective Tension Lbs.	Power each Rope will Transmit H.P.	Centre to centre of ropes Ins.	Diam. of smallest pulley Ins.
½	.1963	.081	47	16	31	4.43	⅞	15
⅝	.3067	.125	72	24	48	6.84	1	18
¾	.4417	.184	106	35	71	10.07	1⅛	22
⅞	.6013	.25	144	48	96	13.67	1 5/16	26
1	.7854	.33	190	63	127	18.05	1½	30
1¼	1.2272	.51	294	98	196	27.90	1 13/16	37
1½	1.7671	.74	426	142	284	40.48	2⅛	45
1¾	2.4053	1.00	576	192	384	54.70	2½	52
2	3.1416	1.30	750	250	500	71.10	2¾	60

Splicing Ropes.—The splicing of ropes is a matter of great importance, and should never be left to unskilled hands. In order to secure a sufficiently strong joint its length requires to be from forty to fifty times the diameter of the rope, or say from 6 to 7 feet for main, and from 4 to 5 feet for secondary drives. The different strands should be neatly interlocked, so that the thickness of the rope is not increased. The strength of a good splice is only about 70 per cent of that of the rope.

Rope Pulleys.—The pulleys used for rope gearing are made with V-shaped grooves around their rims.

The sides of the grooves usually make an angle of 45° with each other. The ropes must never be allowed to rest on the bottom of the grooves, but only on the sides, as shown. They are thus wedged into the grooves, and the resistance to slipping is thereby greatly increased. This has the effect, however, of producing greater wear and tear of the ropes. In the case of guide pulleys the rope should always rest on the bottom of the groove. In the case of belts running on flat-rimmed pulleys, sufficient adhesion to prevent slipping of the belt on the pulleys is obtained by stretching the belt over the pulleys with an initial tension. But with ropes it is found that smoother working, and less wear and tear take place when the rope is put on the pulley with as little initial stress as possible. Hence the reason for increasing the frictional resistance between the rope and its pulleys by allowing the rope to wedge itself into the grooves in the pulley rim.

The necessary pressure between the rope and the sides of the grooves to ensure sufficient frictional adhesion at the commencement of motion, is simply that due to the weight of the hanging parts of the ropes between the pulleys. The pulleys should, therefore, be large, and be placed at a sufficient horizontal distance apart, so as to have the arcs of contact between the rope and the pulleys as great as possible.

Fig. 110 shows the construction of a large rope fly-wheel, in which the wrought-iron bolt, B, connects the grooved rim, G R, with the wheel boss, W B, and thus receives the tensile stress due to centrifugal action on G R.

In horizontal drives, the tight side of the rope should always be in contact with the lower parts of the pulleys, and the slack side above, so as to obtain a maximum arc of contact between the rope and the pulleys. The groove action of rope pulleys also acts in opposition to centrifugal force, which latter is so potent a factor in belt-driving.

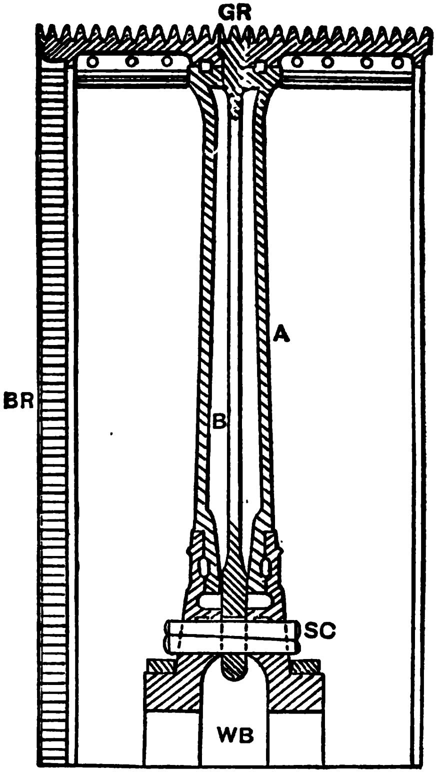


FIG. 110.
SECTION OF LARGE ROPE PULLEY.

Index to parts.—WB, wheel boss; SC, steel cotters; B, wrought iron bolt; A, arm; GR, grooved rim; BR, barring rack.

Ropes can therefore be run at a much higher velocity than belts. Although the usual velocity of rope transmission does not exceed 5,000 feet per minute, yet there are cases where satisfactory results are being obtained from rope transmission at a speed of over 7,000 feet per minute, and the claim is made that the ropes last as long as when running at 5,000 feet per minute. The following formula provides an ample, though not extravagant margin of safety against collapse to individual ropes in fair-sized installations, and also conforms to the regular findings of engineering practice:

$$\text{H. P.} = \frac{V\,(24.6d - 12)}{4860 - 1000\,d},$$

in which H. P. = horsepower transmitted by good 3-strand cotton driving ropes running on pulleys having diameters not less than 30 times the diameter of rope, V = velocity of rope in feet per minute, and d = diameter of rope in inches. (Arc of contact of rope and pulleys assumed as 170°.)

With regard to the relative cost of ropes and of belts, practical experience has shown a very great advantage in favor of the former, and the advantage is more marked in the case of thin ropes, say 13/16-in. in diameter, such as are used for driving light machinery, like ring spinning-frames.

Besides the cases mentioned, there are many other directions in which rope driving may be adopted with advantage, particularly where the electrical transmission of power is concerned, for coal-washing machinery, for colliery fans, brick-making machinery, and the like

Casing of Pulleys.—An important consideration often omitted when arranging for rope driving is, casing the sides of pulleys with sheet-metal or boards, without which the displacement of air by the arms of large high-speed pulleys adds materially to the profitless load.

Minimum Diameter of Smallest Pulley.—The item which appears most naturally to offer itself first for consideration, is the bending, and consequent gripping capability of the rope itself. Soon after cotton came into use for main driving ropes it was discovered that a three-strand rope of this material would cling firmly to a pulley thirty times its own diameter, but would gradually relax its hold as the relative diameter decreased from this standard. Subsequent practice places the validity of this conclusion beyond the possibility of doubt.

Proportionate Durability of Ropes on Large and Small Circumferences.—Too much importance can scarcely be attached to the relative proportions of pulleys and ropes, for it is absolutely certain that both driving force and durability suffer most materially from the non-observance of the thirty-diameter regulation, and that nothing, not even space restriction, has gone so much against the successful application of ropes to the driving of dynamos, or transmission from high-speed motors, as a persistent effort to get below this natural limitation. Therefore, instead of adding to the burden of the drive by employing an abnormal quantity of thick ropes to make up for grip depreciation when large pulleys prove impracticable, it is far better to divide the total bulk nominally required into the necessary number of smaller ropes.

Arc of Contact.—Before dismissing the consideration of pulley diameters it will be well to notice another phase of the same question—viz., that which deals with the arc of contact. It needs but little reasoning to prove that rope contact reaches its maximum when both pulleys are equal in size; but when unequal the utmost quantity realizable is that carried by the smallest pulley, whether it be driver or driven.

Trailing Span of Ropes.—The arc of contact is also affected by the position occupied by the trailing span of the ropes, which, when above the pulleys, undoubtedly gives the maximum bearing surface. This position is not always attainable, nor is it under all circumstances desirable. For example, an erratic delivery of power or variable load may set up oscillations in the responsive ropes of such violence as to fling them entirely off the pulleys. Now if the direction of rotation is so arranged as to bring the trailing span below the pulleys, any tendency on the part of the ropes to wander from their appointed track is held in check by the pull on the right side, while gravitation tends to control the vibrations on the slack side.

Loss due to reverse driving may for safety be calculated as the difference between the average of 170° and the arc of actual contact, which is usually measured from the tangent point of a curve whose depth (from a straight line drawn along the bottom of the pulleys) is one-tenth that of the distance from center to center of the shafts and the taut portion of the rope stretched across the top of the pulleys. With oblique driving the same tendencies do not prevail.

Angular Driving.—There is no class of power transmission which lends itself better to the treatment of what may be considered awkward conditions than rope driving. With the aid of guide pulleys any angle may be negotiated or corner turned.

Shafts Slightly Out of Alignment.—When shafts are only slightly out of alignment ropes will direct themselves without the aid of guide pulleys, but to what extent this angularity may be carried depends largely upon center distances and pulley diameters. For example, under what may be considered good conditions —say with centers not less than three times the average diameter of the pulleys—a deflection of 7° might be ventured upon. Should the angle of deflection from the parallel prove too great to be treated in the above mentioned manner, the drive may be split up into two or more spans by the introduction of intermediate shafts and the angle divided between them. An angle of 15° was thus successfully negotiated. It should be stated that the grooves in all the pulleys were turned to an angle of 50°, and rather more rope applied than would be considered necessary under ordinary conditions.

Grooves.—The fact that when loads are constant the trailing span curves by its own weight down upon or away from the pulleys, according to the direction of rotation, and is ejected, so to speak, without any disturbance or apparent effort, also that a cotton rope may be pressed as tightly as possible round the entire circumference, and will yet fall away immediately the ends are liberated, pretty conclusively demonstrates the futility of seeking easement in curvatures to match in some measure the roundness of the rope itself.

Angular Grooves.—Curves being disallowed, there remains only the angular groove. Undoubtedly, every rope has its most suitable groove angle, but as the providing of a range of templates to agree with every advance in size would, perhaps, prove more inconvenient than advantageous, two leading angles have been fixed upon for all driving ropes, viz., 40° for those above 1¼ ins. in diameter, and 30° below that size; excepting, however, the small bands used for driving cotton machinery—say, ½ in. in diameter and even less, when angles as sharp as 15° are brought into requisition. In setting out a groove it should always be borne in mind that a rope is to all intents and purposes an elastic wedge, and reaches its highest driving capabilities when it is compressed to the shape of the groove itself. This form ought, therefore, always to be anticipated, for if the sides of the groove are merely shown as tangents to the circle representing a given diameter of rope, there is always the danger of making the groove too shallow, so that when compression takes place the rope is liable to find its way to the bottom, or, what is almost as detrimental, work upon the semi-circle with which grooves generally terminate, when grip is considerably weakened and slipping supervenes, the true indication of which is heated pulley-rims.

Flangeless Grooves.—Finding that superposed flanges do not contribute in any way to better transmission, but rather prevent the rope falling direct into its working position if the driving is at all unsteady, besides crippling any extension in the direction of a larger diameter, many manufacturers are now adopting flangeless grooves, which are set out in precisely the same manner, only that the angles are extended and simply rounded off at the terminals.

Driving Ropes.—Many and various are the materials which have been pressed into the service of the ropemaker—an almost infinite variety of fibers, through the whole range of hemps and flaxes, from the coarse-grained manila to the silk-like ramie, and finally cotton.

Manila.—Owing, it may be conjectured, to the fact that rope driving was cradled in what may be styled the home of the manila trade, that material gained for a time precedence above all others, and, if merely judged on the lines of resistance to tensile strains, would still hold the field. The fibers are, however, of so harsh and wiry a nature that they cannot be compressed into a solid mass by the successive twisting operations to which they are subjected, but, instead of forming a perfect impact, the strands merely rest upon each other, thus constructing what may be termed a spiral tube. The strands therefore scrub one upon another with every bend of the rope until the center is reduced to a fine powder.

The permanent set is thus quickly reached, while gripping power is so limited that any reduction in the pulleys from 45 diameters must be reciprocated with a corresponding increase in the amount of rope.

Construction of Ropes.—Much depends upon the construction of a driving rope. Fibrous ropes are held together by a series of twisting operations from the fiber to the yarns, and then the strands, which are locked together by a reversal of the same agency. Ropes of three, four, and even seven strands are used.

A study of a section of three-strand rope discloses, among other advantages, the effect of triangulated strains upon pliable material under the equal pressure exerted by the twisting operation already mentioned. It is in fact a trinity of elastic, spiral wedges, so impacted that no space remains unoccupied, and so supple that they will bend to and fro without disturbing their mutual relationship. This venerable principle as applied to rope making, holds as good to-day as ever, and so far admits of no improvement, except probably in a few minor details.

Lubrication of Driving Ropes.—The chief function of lubricating material as applied to cotton driving ropes, is to provide a smooth, filmy coating to assist the lay, and prevent friction between the twisted surfaces in contact where the strands impact with each other. The lubricant should, however, be applied sparingly, and only at long intervals, unless the ropes are working in an unusually dry atmosphere. Greasy compounds should be avoided, as they will tend to induce slipping. When ropes are inclined to slip, due to an over-dose of grease, a liberal application of whiting will absorb the grease, and then fall away in flakes.

A good lubricant for rope drives is composed of saponified tallow, wax and plumbago.

WHEEL GEARING.

General.—Toothed wheels are employed where motion has to be transmitted with accuracy. The teeth should be formed so that the friction between them shall be the least possible. The diameter of the wheels is measured at the pitch circle. The distance from the center to center of the teeth measured along the pitch circle is called the circular pitch, or commonly the "pitch." If D be the diameter of the pitch circle of a wheel, N the number of teeth, and P the pitch of the teeth, then $D \times 3.1416 = N \times P$. The "diametral" pitch of a wheel (used chiefly in smaller sizes) is the length bearing the same proportion to the circular pitch that the diameter of a circle bears to the circumference. Therefore if P is the circular pitch, p diametral pitch, D diameter of pitch circle, and N the number of teeth, we have:

Circular pitch........... $= P = \dfrac{3.1416\,D}{N} = 3.1416\,p$

Diametral pitch......... $= p = \dfrac{P}{3.1416} = \dfrac{D}{N}\,0.318\,P$

$= 0.318\,P$

Diameter of pitch circle.. $=D=N\,p=\dfrac{P\times N}{3.1416}=0.318PN$

Number of teeth........ $=N=\dfrac{3.1416\,D}{P}=\dfrac{D}{p}$

The following table gives the diametral pitches corresponding to circular pitches from ½ in. to 12 in.:

TABLE 28.

Circular Pitch. in.	Diametral Pitch.	Circular Pitch. in.	Diametral Pitch.	Circular Pitch. in.	Diametral Pitch.
½	0.1592	2¼	0.7162	5	1.5916
⅝	0.1989	2½	0.7958	5½	1.7507
¾	0.2387	2¾	0.8754	6	1.9099
⅞	0.2786	3	0.9549	6½	2.0691
1	0.3183	3¼	1.0345	7	2.2283
1⅛	0.3581	3½	1.1141	7½	2.3875
1¼	0.3979	3¾	1.1936	8	2.5465
1⅜	0.4377	4	1.2732	9	2.8647
1½	0.4775	4¼	1.3528	10	3.1829
1¾	0.5570	4½	1.4324	11	3.5014
2	0.6366	4¾	1.5120	12	3.8200

The so-called diametral pitch adopted by Messrs. Brown and Sharpe, is the reciprocal of the foregoing, or $\dfrac{1}{p}$.

For this method of computing the diametral pitch, the following tables will be of service. The first gives the equivalent diametral pitches for circular pitches of from 2 to $\frac{1}{16}$ in.; and the second the equivalent circular pitches for diametral pitches of from ¾ to 60.

TABLE 29.

Circular Pitch. in.	Diametral Pitch.	Circular Pitch. in.	Diametral Pitch.	Circular Pitch. in.	Diametral Pitch.
2	1.5708	1	3.1416	1/3	9.4248
1⅞	1.6755	15/16	3.3510	5/16	10.0531
1¾	1.7952	⅞	3.5904	1/4	12.5664
1⅝	1.9333	13/16	3.8666	2/9	14.1372
1½	2.0944	¾	4.1888	1/5	15.7080
1 7/16	2.1855	11/16	4.5696	3/16	16.7552
1⅜	2.2848	⅔	4.7124	1/6	18.8496
1 5/16	2.3936	⅝	5.0265	1/7	21.9911
1¼	2.5133	9/16	5.5851	1/8	25.1327
1 3/16	2.6456	½	6.2832	1/9	28.2743
1⅛	2.7925	7/16	7.1808	1/10	31.4159
1 1/16	2.9568	⅜	8.3776	1/16	50.2655

TABLE 30.

Diametral Pitch.	Circular Pitch. in.	Diametral Pitch.	Circular Pitch. in.	Diametral Pitch.	Circular Pitch. in.
¾	4.1888	8	0.3927	26	0.1208
1	3.1416	9	0.3491	28	0.1122
1¼	2.5133	10	0.3142	30	0.1047
1½	2.0944	11	0.2856	32	0.0982
1¾	1.7952	12	0.2618	34	0.0924
2	1.5708	13	0.2417	36	0.0873
2¼	1.3963	14	0.2244	38	0.0827
2½	1.2566	15	0.2094	40	0.0785
2¾	1.1424	16	0.1963	42	0.0748
3	1.0472	17	0.1848	44	0.0714
3½	0.8976	18	0.1745	46	0.0683
4	0.7854	19	0.1653	48	0.0654
5	0.6283	20	0.1571	50	0.0628
6	0.5236	22	0.1428	56	0.0561
7	0.4488	24	0.1309	60	0.0524

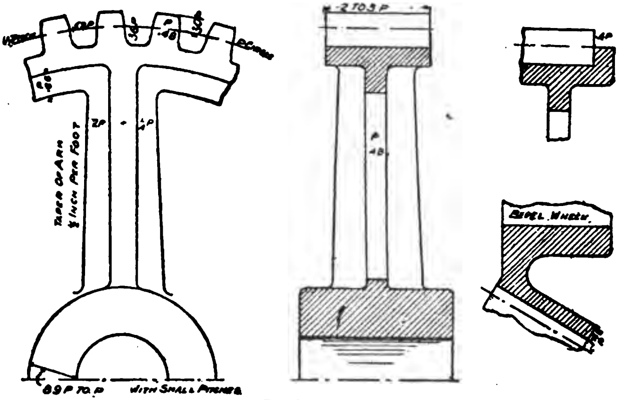


FIG. 111.

Proportions of Teeth.—The ordinary proportions of the teeth and arms of spur and bevel wheels in terms of the pitch are given in Fig. 111. In this case:

Thickness of teeth = 0.48 pitch.

Space between teeth = 0.52 pitch.

Length of teeth from pitch line to point = 0.3 pitch.

Length of tooth from pitch line to bottom of tooth = 0.4 pitch.

The following proportions are given by Unwin as preferable to the foregoing:

Thickness of tooth. . . . = 0.47 P — 0.02 to 0.48 P — 0.03
Width of space. . . . . . . = 0.53 P + 0.02 to 0.52 P + 0.03
Side clearance . . . . = 0.06 P + 0.04 to 0.04 P + 0.06
Height above pitch line. = 0.3 P to 0.35 P
Depth below pitch line. = 0.35 P + 0.08 to 0.4 P + 0.08
Total height of teeth. . = 0.65 P + 0.08 to 0.75 P + 0.08

The length of teeth is a matter in which much variation exists in practice, the present tendency being to adopt shorter teeth than in the older practice. Though this diminishes wearing surface, the teeth are less liable to fracture, which is an important point in high-speeded and heavily-stressed wheels. The old proportions for depth of teeth, including bottom clearance, gave 12/15 of the pitch. With high-speed wheels doing heavy work, this is too much. The proportion, 7/10, is better. During recent years several engineers have recommended as a general rule for heavy mill wheels, that the depth should not exceed one-half the pitch.

Helical Wheels.—Helical wheels are a legitimate development of the stepped gears, the object of which is to diminish the sliding action of the teeth. Thus in a stepped gear the total arc of contact, which measures the amount of sliding, is divided between several short teeth, and the sliding on each is reduced to

$$\frac{\text{Arc of contact}}{\text{number of teeth}}.$$

A helical tooth is simply one in which the steps merge into a spiral line, and as the points of contact change continually along the entire tooth plank, there is no sliding at all, but simply rolling. The double helical teeth are employed in order to balance the lateral forces which exercise undue pressure, tending to force the teeth away from contact in single helical wheels.

Spiral Gears.—Spiral gears bear a family resemblance to the helical gears. The difference lies in the direction of their axes. They are both alike screw-gears, and must be so designed. But while the action of helical wheels is entirely rolling, that of screw-gears is wholly sliding. The difference is due to the different positions of the wheel axis, which in the first case are parallel, and in the second at right, or other angles.

Worm-Gears.—Worm-Gears are particular applications of screw-gears. The worm is a wheel having a single thread, or tooth of very short axial pitch. The teeth of the wheel in which it gears are short sections of threads of extremely long axial pitch.

As in screw-gears, the action is wholly sliding. Double and treble threaded worms contain two and three screw threads. The axial pitch of the wheel teeth is much shorter, and the angle, or slope, greater than in those which gear with a single threaded worm. That the wheel teeth are short lengths of spirals of very long axial pitch is clearly evident in the case of straight wheel teeth of uniform section set diagonally on the rim, but is not so clear in the case of teeth which are cut to envelop the worm tooth. The latter is the proper form for the best results, especially when heavy gearing is in question, though light wheels of small pitch may be formed by the first method.

Wheel Arms.—The shapes of arms of wheels are settled more by convenience of manufacture than by any hard and fast rules. The strength is generally much in excess of that of the teeth, in order to absorb vibration. The arms are of T or ⌶, or 0 sections, the latter being chiefly used on change gears. T and + are employed mostly in pattern wheels, and ⌶ in those made by machine, while the simple disc is reserved for many wheels in small diameter.

Number of Arms of Spur Wheels.—Under 3 ft. diameter should be 4; from 3 ft. to 8 ft., 6 arms; from over 8 ft. to 16 ft., 8 arms; from over 16 ft. to 24 ft., 10 arms.

Wheel Bosses.—Not the least important part of a wheel is its boss. These often cause fracture in consequence of being too large. The boss should never be heavy, especially when adjacent arms are light, and extra metal around keys should be provided.

Strength of Spur-wheel Teeth.—Innumerable rules have been formulated for the strength of the teeth of spur wheels, many of which give widely different results. Much of the existing confusion is due not only to the various estimates of the strength of the material, and the allowable factor of safety, but also to the manner in which the load is taken to fall on the teeth. It is frequently assumed that the tooth is liable to fracture across the corner, in which case only a portion of the breadth of the tooth equal to the length $\times \frac{1}{\sqrt{2}}$ ought to be considered as available. No doubt this condition may be justly assumed with roughly-cast gearing, or when the axes of the wheels are not exactly parallel or the bearings rigidly fixed, as well as in the case of crane gearing, or in gearing subject to shocks or rough usage. For well-made machine-moulded, or cut wheels, such as are now very generally used for main-driving gear, however, the teeth may reasonably be assumed to have a fairly even bearing throughout the whole of their length.

Taking a tooth of ordinary proportions, and making

TABLE 31.—HORSE-POWER OF SPUR WHEELS.
(Per Revolution per Minute and per Inch of Breadth.)

Pitch in Inches.	Diameter of Wheel.																				
	ft in 0 6	ft in 0 9	ft in 1 0	ft in 1 3	ft in 1 6	ft in 1 9	ft in 2 0	ft in 2 3	ft in 2 6	ft in 2 9	ft in 3 0	ft in 3 6	ft in 4 0	ft in 4 6	ft in 5 0	ft in 5 6	ft in 6 0	ft in 7 0	ft in 8 0	ft in 9 0	ft in 10 0
1	.006	.009	.012	.015	.018	.021	.024	.027	.030	.033	.036	.042	.048	.054	.060	.066	.072	.084	.096	.108	.120
1¼	.007	.011	.015	.019	.022	.026	.030	.034	.037	.041	.045	.052	.060	.067	.075	.082	.090	.105	.120	.135	.150
1½	.009	.013	.018	.023	.027	.031	.036	.040	.045	.050	.054	.063	.072	.081	.090	.099	.108	.126	.144	.162	.180
1¾	.010	.015	.021	.026	.032	.036	.042	.047	.052	.058	.063	.073	.084	.094	.105	.115	.126	.147	.168	.189	.210
2	.012	.018	.024	.030	.036	.042	.048	.054	.060	.066	.072	.084	.096	.108	.120	.132	.144	.168	.192	.216	.240
2¼	....	....	.027	.034	.040	.047	.054	.061	.067	.073	.081	.094	.108	.121	.135	.148	.162	.189	.216	.243	.270
2½	....	....	.030	.038	.044	.052	.062	.068	.074	.082	.090	.104	.120	.134	.150	.164	.180	.210	.240	.270	.300
2¾	....	....	....	.041	.049	.053	.066	.074	.082	.091	.099	.115	.132	.148	.165	.181	.198	.231	.264	.297	.330
3	....	....	....	.045	.054	.063	.072	.081	.090	.099	.108	.126	.144	.162	.180	.198	.216	.252	.288	.324	.360
3¼	....	....	....	.049	.058	.068	.078	.088	.097	.107	.117	.136	.156	.175	.195	.214	.234	.273	.312	.350	.390
3½	....	....	....	....	.064	.072	.084	.094	.104	.116	.126	.146	.168	.188	.210	.230	.252	.294	.336	.378	.420
3¾	....	....	....	....	.067	.078	.090	.101	.112	.124	.135	.157	.180	.202	.225	.247	.270	.315	.360	.405	.450
4	....	....	....	....	....	.084	.096	.108	.120	.132	.144	.168	.192	.216	.240	.264	.288	.336	.384	.432	.480
4¼	....	....	....	....	....	.089	.102	.115	.127	.140	.153	.178	.204	.230	.255	.280	.306	.357	.408	.460	.510
4½	....	....	....	....	....	.094	.108	.122	.134	.148	.162	.188	.216	.242	.270	.296	.324	.378	.432	.486	.540
4¾	....	....	....	....	....	....	.114	.128	.142	.157	.171	.199	.228	.256	.285	.313	.342	.399	.455	.512	.570
5	....	....	....	....	....	....	.120	.135	.150	.165	.180	.210	.240	.270	.300	.330	.360	.420	.480	.540	.600
5½	....	....	....	....	....	....	....	.148	.164	.182	.198	.230	.264	.296	.330	.362	.396	.462	.528	.594	.660
6	....	....	....	....	....	....	....	.162	.180	.198	.216	.252	.288	.324	.360	.396	.432	.504	.576	.648	.720

a suitable allowance for wear, we have, if the stress on the material is taken at about 1,500 lbs. per square inch,

$$\text{H. P.} = \frac{\text{D p b N}}{1000},$$

in which H. P. = horsepower; D the diameter of the wheel in inches; p the pitch; b the breadth; and N the number of revolutions per minute. It is assumed that one-half the load falls on each tooth, and that the teeth bear uniformly throughout their whole breadth. Table 31, based on these assumptions, gives the horsepower per revolution and per inch of width of spur wheels.

Weight of Spur-wheel Castings.—

$$W = \frac{P^2 \times N \times B}{C},$$

where P = pitch in inches; N = number of teeth; B = breadth in inches; W = weight in cwt.; C = constant, which up to 2 in. pitch = 270; from 2 in. to 3 in. = 250; from 3 in. to 4 in. = 245. Roughly, the weight of an iron casting equals fourteen times the weight of pattern.

Wheel-gearing Constants.—Table 32 of constants (C) will be found of service in calculations involving the pitch P, number of teeth N, and radius of pitch circle R, of spur wheels.

$$R = C \times P. \qquad P = \frac{R}{C}. \qquad C = \frac{R}{P}.$$

Examples.—Find the radius of pitch circle of a spur wheel having 45 teeth 1¾-in. pitch.

At the intersection of lines 40 and 5 in Table 32, we find the constant 7.16. Then $7.16 \times 1.75 = 12.53$, or 25 in. diameter.

TABLE 32.—TABLE OF WHEEL-GEARING CONSTANTS.

N	0	1	2	3	4	5	6	7	8	9
0	0.00	0.159	0.318	0.477	0.637	0.796	0.955	1.114	1.273	1.43
10	1.39	1.75	1.91	2.07	2.23	2.39	2.55	2.71	2.86	3.02
20	3.18	3.34	3.50	3.66	3.82	3.98	4.14	4.30	4.46	4.62
30	4.77	4.93	5.09	5.25	5.41	5.57	5.73	5.89	6.05	6.21
40	6.37	6.53	6.68	6.84	7.00	7.16	7.32	7.48	7.64	7.80
50	7.96	8.12	8.28	8.44	8.59	8.75	8.91	9.07	9.23	9.39
60	9.55	9.71	9.87	10.03	10.19	10.35	10.50	10.66	10.82	10.98
70	11.14	11.30	11.46	11.62	11.78	11.94	12.10	12.25	12.41	12.57
80	12.73	12.89	13.05	13.21	13.37	13.53	13.69	13.85	14.01	14.16
90	14.32	14.48	14.64	14.80	14.96	15.12	15.28	15.44	15.60	15.76
100	15.92	16.07	16.23	16.39	16.55	16.71	16.87	17.03	17.19	17.35
110	17.51	17.67	17.83	17.98	18.14	18.30	18.46	18.62	18.78	18.94
120	19.10	19.26	19.42	19.58	19.73	19.89	20.05	20.21	20.37	20.53
130	20.69	20.85	21.01	21.17	21.33	21.49	21.65	21.80	21.96	22.12
140	22.28	22.44	22.60	22.76	22.92	23.08	23.24	23.40	23.55	23.71
150	23.87	24.03	24.19	24.35	24.51	24.67	24.83	24.99	25.15	25.31
160	25.46	25.62	25.78	25.94	26.10	26.26	26.42	26.58	26.74	26.90
170	27.06	27.21	27.37	27.53	27.69	27.85	28.01	28.17	28.33	28.49
180	28.65	28.81	28.97	29.13	29.28	29.44	29.60	29.76	29.92	30.08
190	30.24	30.40	30.56	30.72	30.88	31.04	31.19	31.35	31.51	31.67
200	31.83	31.99	32.15	32.31	32.47	32.63	32.79	32.95	33.10	33.26
210	33.42	33.58	33.74	33.90	34.06	34.22	34.38	34.54	34.70	34.85
220	35.01	35.17	35.33	35.49	35.65	35.81	35.97	36.13	36.29	36.45
230	36.61	36.76	36.92	37.08	37.24	37.40	37.56	37.72	37.88	38.04
240	38.20	38.36	38.51	38.67	38.83	38.99	39.15	39.31	39.47	39.63
250	39.79	39.95	40.11	40.27	40.42	40.58	40.74	40.90	41.06	41.22
260	41.38	41.54	41.70	41.86	42.02	42.18	42.34	42.49	42.65	42.81
270	42.97	43.13	43.29	43.45	43.61	43.77	43.93	44.09	44.25	44.40
280	44.56	44.72	44.88	45.04	45.20	45.36	45.52	45.68	45.84	46.00
290	46.15	46.31	46.47	46.63	46.79	46.95	47.11	47.27	47.43	47.59

Find pitch of teeth if $R = 20$ and $N = 93$. The constant for $93 = 14.8$ and $P = \frac{R}{C} = \frac{20}{14.8} = 1.35$ in.

Find number of teeth when $R = 32$ and $P = 2$ in.

We have $C = \frac{R}{P} = \frac{32}{2} = 16$, and reference to the table of constants shows that the nearest number corresponding to this is 16.07 at the intersection of 100 and 1 and therefore $N = 101$ teeth.

TABLE 33.—CENTRIFUGAL FORCE IN LB. OF A WEIGHT OF 1 LB. AT VARIOUS SPEEDS.

Dia. of Path	Revolutions per Minute.								
In.	100	200	300	400	500	600	700	800	900
12	1.70	6.81	15.32	27.24	42.56	61.3	83.43	108.97	137.93
18	2.55	10.22	22.98	40.86	63.85	91.95	125.16	163.47	206.90
24	3.41	13.63	30.65	54.49	85.13	122.60	166.87	217.95	275.86
30	4.26	17.03	38.31	68.56	106.42	153.26	208.60	272.42	344.80
36	5.11	20.44	45.96	81.73	127.70	183.90	250.32	326.94	413.80
42	5.96	23.84	53.63	95.35	149.00	214.55	292.00	381.40	482.70
48	6.81	27.25	61.29	108.97	170.26	245.20	333.74	435.90	551.72
54	7.66	30.66	68.96	122.60	191.57	275.88	375.45	490.40	620.65
60	8.51	34.06	76.61	137.12	212.84	306.52	417.20	544.84	689.60
66	9.36	37.47	84.28	149.85	234.13	337.16	458.90	599.40	758.70
72	10.21	40.88	91.94	163.45	255.39	367.80	500.61	653.85	837.58

Dia. of Path	Revolutions per Minute.								
In.	1000	1100	1200	1400	1600	1800	2000	2500	3000
12	170.25	206.00	245.15	333.75	435.90	551.65	681.00	1064.00	1532.25
18	255.40	309.0	367.7	500.6	653.8	827.4	1021.6	1596.2	2298.8
24	340.50	412.0	490.3	667.5	871.8	1103.3	1362.0	2128.0	3064.5
30	425.60	514.9	612.9	834.4	1089.8	1379.0	1543.6	2660.2	3831.05
36	510.80	618.0	735.5	1001.3	1307.6	1654.9	2043.0	3192.0	4596.7
42	595.90	721.0	858.0	1168.2	1525.8	1931.0	2383.4	3724.0	5362.8
48	681.00	824.0	980.6	1335.0	1743.6	2206.6	2724.0	4256.0	6129.0
54	766.20	927.0	1103.3	1502.2	1961.5	2482.8	3065.0	4789.0	6897.0
60	852.00	1039.9	1235.8	1768.8	2179.6	2758.0	3087.2	5320.4	7662.1
66	936.40	1133.2	1348.5	1835.8	2397.7	3034.2	3745.5	5852.5	8427.8
72	1021.50	1236.0	1470.9	2002.5	2615.2	3309.9	4086.0	6384.0	9193.5

TABLE 34.

SQUARES, CUBES, SQUARE ROOTS AND CUBE ROOTS.

No.	Square	Cube.	Square Root. √	Cube Root. ∛	No.	Square	Cube.	Square Root. √	Cube Root ∛
1	1	1	1·000	1·000	51	2601	132651	7·141	3·708
2	4	8	1·414	1·259	52	2704	140608	7·211	3·732
3	9	27	1·732	1·442	53	2809	148877	7·280	3·756
4	16	64	2·000	1·587	54	2916	157464	7·348	3·779
5	25	125	2·236	1·709	55	3025	166375	7·416	3·802
6	36	216	2·449	1·817	56	3136	175616	7·483	3·825
7	49	343	2·645	1·912	57	3249	185193	7·549	3·848
8	64	512	2·828	2·000	58	3364	195112	7·615	3·870
9	81	729	3·000	2·080	59	3481	205379	7·681	3·892
10	100	1000	3·162	2·154	60	3600	216000	7·745	3·914
11	121	1331	3·316	2·223	61	3721	226981	7·810	3·936
12	144	1728	3·464	2·289	62	3844	238328	7·874	3·957
13	169	2197	3·605	2·351	63	3969	250047	7·937	3·979
14	196	2744	3·741	2·410	64	4096	262144	8·000	4·000
15	225	3375	3·872	2·466	65	4225	274625	8·062	4·020
16	256	4096	4·000	2·519	66	4356	287496	8·124	4·041
17	289	4913	4·123	2·571	67	4489	300763	8·185	4·061
18	324	5832	4·242	2·620	68	4624	314432	8·246	4·081
19	361	6859	4·358	2·668	69	4761	328509	8·306	4·101
20	400	8000	4·472	2·714	70	4900	343000	8·366	4·121
21	441	9261	4·582	2·758	71	5041	357911	8·426	4·140
22	484	10648	4·690	2·802	72	5184	373248	8·485	4·160
23	529	12167	4·795	2·843	73	5329	389017	8·544	4·179
24	576	13824	4·898	2·884	74	5476	405224	8·602	4·198
25	625	15625	5·000	2·924	75	5625	421875	8·660	4·217
26	676	17576	5·099	2·962	76	5776	438976	8·717	4·235
27	729	19683	5·196	3·000	77	5929	456333	8·774	4·254
28	784	21952	5·291	3·036	78	6084	474552	8·831	4·272
29	841	24389	5·385	3·072	79	6241	493039	8·888	4·290
30	900	27000	5·477	3·107	80	6400	512000	8·944	4·308
31	961	29791	5·567	3·141	81	6561	531441	9·000	4·326
32	1024	32768	5·656	3·174	82	6724	551368	9·055	4·344
33	1089	35937	5·744	3·207	83	6889	671787	9·110	4·362
34	1156	39304	5·830	3·239	84	7056	592704	9·165	4·379
35	1225	42875	5·916	3·271	85	7225	614125	9·219	4·396
36	1296	46656	6·000	3·301	86	7396	636056	9·273	4·414
37	1369	50653	6·082	3·332	87	7569	658503	9·327	4·431
38	1444	54872	6·164	3·361	88	7744	681472	9·380	4·447
39	1521	59319	6·244	3·391	89	7921	704969	9·433	4·464
40	1600	64000	6·324	3·420	90	8100	729000	9·486	4·480
41	1681	68921	6·403	3·448	91	8281	753571	9·539	4·497
42	1764	74088	6·480	3·476	92	8464	778688	9·591	4·514
43	1849	79507	6·557	3·503	93	8649	804357	9·643	4·530
44	1936	85184	6·633	3·530	94	8836	830584	9·695	4·546
45	2025	91125	6·708	3·556	95	9025	857375	9·746	4·560
46	2116	97336	6·782	3·583	96	9216	884736	9·797	4·578
47	2209	103823	6·855	3·608	97	9409	912673	9·848	4·594
48	2304	110592	6·928	3·634	98	9604	941192	9·899	4·610
49	2401	117649	7·000	3·659	99	9801	970299	9·949	4·626
50	2500	125000	7·071	3·684	100	10000	1000000	10·000	4·641

TABLE 34.—CONTINUED.

SQUARES, CUBES, SQUARE ROOTS AND CUBE ROOTS.

No.	Square.	Cube.	Square Root. √	Cube Root. ∛	No.	Square	Cube.	Square Root. √	Cube Root. ∛
105	11025	1157625	10·24	4·717	360	129600	46656000	18·97	7·113
110	12100	1331000	10·48	4·791	365	133225	48627125	19·10	7·146
115	13225	1520875	10·72	4·862	370	136900	50653000	19·23	7·179
120	14400	1728000	10·95	4·932	375	140625	52734375	19·36	7·211
125	15625	1953125	11·18	5·000	380	144400	54872000	19·49	7·243
130	16900	2197000	11·40	5·065	385	148225	67066625	19·62	7·274
135	18225	2460375	11·61	5·129	390	152100	59319000	19·74	7·306
140	19600	2744000	11·83	5·192	395	156025	61629875	19·87	7·337
145	21025	3048625	12·04	5·253	400	160000	64000000	20·00	7·368
150	22500	3375000	12·24	5·313	405	164025	66430125	20·12	7·398
155	24025	3723875	12·44	5·371	410	168100	68921000	20·24	7·428
160	25600	4096000	12·64	5·428	415	172225	71473375	20·37	7·459
165	27225	4492125	12·84	5·484	420	176400	74088000	20·49	7·488
170	28900	4913000	13·03	5·539	425	180625	76765625	20·61	7·518
175	30625	5359375	13·22	5·593	430	184900	79507000	20·736	7·547
180	32400	5832000	13·41	5·646	435	189225	82312875	20·856	7·576
185	34225	6331625	13·60	5·698	440	193600	85184000	20·976	7·605
190	36100	6859000	13·78	5·748	445	198025	88121125	21·095	7·634
195	38025	7414875	13·96	5·798	450	202500	91125000	21·213	7·663
200	40000	8000000	14·14	5·848	455	207025	94196375	21·330	7·691
205	42025	8615125	14·31	5·896	460	211600	97336000	21·447	7·719
210	44100	9261000	14·49	5·943	465	216225	100544625	21·563	7·747
215	46225	9938375	14·66	5·990	470	220900	103823000	21·679	7·774
220	48400	10648000	14·83	6·036	475	225625	107171875	21·794	7·802
225	50625	11390625	15·00	6·082	480	230400	110592000	21·908	7·829
230	52900	12167000	15·16	6·126	485	235225	114084125	22·022	7·856
235	55225	12977875	15·32	6·171	490	240100	117649000	22·135	7·883
240	57600	13824000	15·49	6·214	495	245025	121287375	22·248	7·910
245	60025	14706125	15·65	6·257	500	250000	125000000	22·360	7·937
250	62500	15625000	15·81	6·299	510	260100	132651000	22·583	7·989
255	65025	16581375	15·96	6·341	520	270400	140608000	22·803	8·041
260	67600	17576000	16·12	6·382	540	291600	157464000	23·237	8·143
265	70225	18609625	16·27	6·423	560	313600	175616000	23·664	8·247
270	72900	19683000	16·43	6·463	580	336400	195112000	24·083	8·339
275	75625	20796875	16·58	6·502	600	360000	216000000	24·494	8·434
280	78400	21952000	16·73	6·542	625	390625	244140625	25·000	8·549
285	81225	23149125	16·88	6·580	650	422500	274625000	25·495	8·662
290	84100	24389000	17·02	6·619	675	455625	307546875	25·980	8·772
295	87025	25672375	17·17	6·656	700	490000	343000000	26·457	8·879
300	90000	27000000	17·32	6·694	725	525625	381078125	26·925	8·983
305	93025	28372625	17·46	6·731	750	562500	421875000	27·386	9·085
310	96100	29791000	17·60	6·767	775	600625	465484375	27·838	9·185
315	99225	31255875	17·74	6·804	800	640000	512000000	28·284	9·283
320	102400	32768000	17·88	6·839	825	680625	561515625	28·722	9·378
325	105625	34328125	18·02	6·875	850	722500	614125000	29·154	9·472
330	108900	35937000	18·16	6·910	875	765625	669921875	29·580	9·564
335	112225	37595375	18·30	6·945	900	810000	729000000	30·000	9·654
340	115600	39304000	18·43	6·979	925	855625	791453125	30·413	9·743
345	119025	41063625	18·57	7·013	950	902500	857375000	30·822	9·830
350	122500	42875000	18·70	7·047	975	950625	926859375	31·224	9·915
355	126025	44738875	18·84	7·080	1000	1000000	1000000000	31·622	10·000

SYSTEMS OF LUBRICATION.

Chain or Ring System.—The main defect in this method of lubrication is that through neglect the oil is liable to become loaded with dirt and foreign matter to such an extent that it ceases to lubricate the bearings unless it is all drained out, and fresh oil put in every week or two. If this is done, the bearings may be fairly well lubricated by the use of this system.

The following conclusions concerning lubrication are the result of a series of experiments on friction and lubrication as recorded by Prof. Jamieson in Vol. two of "Applied Mechanics." Tables A and B are very valuable for reference purposes.

Prof. Jamieson says: "The oil bath probably represents the most perfect lubrication possible, and the limit beyond which friction cannot be reduced by lubrication. The experiments show, that with speeds of from 100 to 200 feet per minute, by properly proportioning the bearing surface to the load, it is possible to reduce the co-efficient of friction as low as .0001. A co-efficient of .005 is easily attainable and probably is frequently attained in ordinary engine bearings, in which the direction of the force is rapidly alternating, and the oil is given an opportunity to get between the rubbing surfaces, while the duration of the force in one direction is not sufficient to allow time for the oil-film to be squeezed out. The extent to which the friction depends on the quantity of the lubrication is shown in a remarkable manner by the following table, which proves that the lubrication can be so diminished that the friction is seven times greater than it was in the oil bath, and yet that the bearing will run without seizing."

COMPARISON OF THE FRICTION WITH THE DIFFERENT METHODS OF LUBRICATION, UNDER AS NEARLY AS POSSIBLE THE SAME CIRCUMSTANCES. LUBRICANT RAPE OIL, SPEED 150 REVOLUTIONS PER MINUTE.

Mode of Lubrication.	Actual Load Lbs. per sq. in.	Coefficient of Friction.	Comparative Friction.
Oil-bath	263	.00139	1
Syphon lubricator	252	.00980	7.06
Pad under journal	272	.00900	6.48

TABLE A.

Observations on the behavior of the apparatus gave reason to believe that with perfect lubrication the speed of minimum friction was from 100 to 150 feet per minute; and that this speed of minimum friction tended to be higher with an increase of load, and also with less perfect lubrication. By the speed of minimum friction is meant, that speed in approaching which, from rest, the friction diminishes, and above which the friction increases.

The following table gives the means of the actual frictional resistances at the surface of the journal per square inch of bearing, at a speed of 300 revolutions per minute, with all nominal loads from 100 lbs. per square inch up to 310 lbs. per square inch.

They also represent the relative thickness or body of the various oils, and (in their order, though perhaps not exactly in their numerical proportions) their relative weight-carrying power. Thus sperm oil, which has the highest lubricating power, has the least weight-carrying power; and though the best oil for light loads, would be inferior to the thicker oils if heavy pressures or high temperatures were to be encountered.

COMPARISON OF THE FRICTION WITH THE VARIOUS LUBRICANTS TRIED, UNDER AS NEARLY AS POSSIBLE THE SAME CIRCUMSTANCES. TEMPERATURE 90°. LUBRICATION BY OIL-BATH.

Lubricant	Mean Resistance Lb.	Per Cent.
Sperm oil	0.484	100
Rape oil	0.512	106
Mineral oil	0.623	129
Lard oil	0.652	135
Olive oil	0.654	135
Mineral grease	1.048	217

TABLE B.

Grease for Lubrication. While grease is not altogether suitable for lubricating machinery running at high speed, it is well adapted for use on slow moving journals. For elevators, conveyors, and for all journals where there is more or less dirt, grease is desirable for purposes of lubrication.

In case of ordinary grease lubrication, the grease should be contained in a screw grease-cup, and by screwing down the cover of the cup, a portion of the grease is forced through the regular oil passage into the bearing.

Best Lubricants for Various Purposes. According to tests made by Prof. Thurston, lubricants are listed as follows:

Low temperatures.—Light mineral lubricating oils.

Very great pressures, slow speed.—Graphite, soapstone and other solid lubricants.

Heavy pressures with slow speeds.—The above, and lard, tallow and other greases.

Heavy pressures and high speeds.—Sperm oil, castor oil and heavy mineral oils.

Light pressures and high speeds.—Sperm, refined petroleum, olive, rape, cotton-seed oils.

Ordinary machines.—Lard oil, tallow oil, heavy mineral oils and the heavier vegetable oils.

Steam cylinders.—Heavy mineral oils, lard, tallow.

Watches and other delicate machinery.—Clarified sperm, neat's foot, porpoise, olive, and light mineral oils.

For mixture with mineral oils, sperm is best, lard is much used, olive and cotton-seed are good.

Cold Tests of Oils.—One of the most important features in the selection of lubricating oils is the lowest temperature that the oil will stand, and still remain fluid. The determination of this property of oil is termed the cold test, performed as follows: A small quantity of the oil to be tested is placed in a glass jar, or large-mouthed bottle which is then packed in a mixture of ice and salt until the oil solidifies. The bottle is then removed from the cold pack, and the oil allowed to soften, being stirred constantly, while at the same time a thermometer is inserted, and the temperature at which the oil again runs is noted.

Efficiency of Different Systems of Lubrication.—Friction varies considerably with the method employed to lubricate bearings.

The friction-reducing efficiency of several different modes of applying oil to bearings, is given in the following table:

EFFICIENCY OF DIFFERENT SYSTEMS OF LUBRICATION, AS SHOWN BY THE FRICTION OF THE JOURNALS OF A STEEL SHAFT REVOLVING IN BEARINGS OF GUN-METAL, LUBRICATED WITH THE SAME KIND OF OIL APPLIED IN DIFFERENT WAYS.

System of Lubrication.	Co-efficient of Friction.
Forced lubrication	.003
Oil-bath-lubrication	.004
Splash-lubrication	.005
Saturated oil-pad lubrication	.014
Self-lubricating bearings	.016
Syphon oil-lubricators	.018
Sight-feed lubricators	.020
Needle-lubricators	.023
Intermittent, or ordinary oiling, average best results with	.025
Intermittent or ordinary oiling by hand with an oil-can, average results with.	.050

Efficient lubrication is best obtained by applying the oil through three oil-holes, one being placed at each side and another at the center of the bearing.

Lubricants for the Bearings of the Journals of Shafts.—The viscosity or body of an oil represents its fluidity, and is the measure of its resistance to being squeezed out of a bearing. The greater the viscosity the less is the liability to being squeezed out. The viscosity of oils varies considerably, but at a temperature of 65° Fahr. it is frequently as follows:

	Viscosity Compared with that of Sperm Oil.
Sperm Oil	1.00
Light mineral oil	1.70
Cotton-seed oil	1.90
Olive oil	2.00
Lard oil	2.20
Rape oil	2.30
Neatsfoot oil	2.40
Tallow oil	2.50
Heavy mineral oil	4.00
Castor oil	23.00

The viscosity decreases as the temperature increases up to 220° Fahr., when all these oils have nearly the same viscosity. This shows the importance of keeping bearings cool, especially those using oil of full body.

The best lubricants for the bearings of engines, shafting and machinery, under ordinary working conditions, are generally as follows:

Description of Bearings.	Description of Lubricant. Mixed with Mineral Oil.
For high-speed shafts with a light pressure on the bearings........	Sperm oil.
For low-speed shafts with a light pressure on the bearings..........	Olive oil.
For high-speed shafts with a heavy pressure on the bearings........	Sperm oil and castor oil.
For low-speed shafts with a heavy pressure on the bearings........	Rape oil and lard oil.
For the crank-shafts of large steam-engines	Castor oil.

The nature of the bearing-metal is an important factor in economical lubrication. It was found in the case of a heavily loaded shaft that some bearing-metals required more oil than others to maintain the same cool temperature. The relative consumption of oil by different bearing-metals is in some cases as follows:

	Relative Consumption of Oil for an Equal Temperature.
Cast-iron bearings for the journals of shafts................	1.00
Brass bearings for the journals of shafts..	.90
Gun-metal bearings for the journals of shafts................	.85
Phosphor-bronze bearings for the journals of shafts..........	.80
Antifriction white-metal bearings for the journals of shafts....	.75

The consumption of oil by bearings of hardened cast-steel, for equality of temperature, is generally at least twice as great as that of gun-metal bearings.

Mineral Oil of fine quality is very effective in cleaning metal, and when used in combination with other oils as a lubricant, it tends to maintain bearings clean and free from clogging. Hence, mixtures of fatty oils and mineral oils are generally more efficient for the lubrication of the bearings of the journals of shafts than either of these oils alone. The following mixtures of oils form efficient lubricants for some bearings:

Description of Bearings.	Composition of Lubricants
For high-speed shafts with light pressure on the bearings.	Sperm oil 20 per cent., olive oil 30 per cent., and light mineral oil 50 per cent.
For low-speed shafts with light pressure on the bearings.	Olive oil 50 per cent., and light mineral oil 50 per cent.
For high-speed shafts with heavy pressure on the bearings.	Lard oil 35 per cent., sperm oil 25 per cent., and heavy mineral oil 40 per cent.
For low-speed shafts with heavy pressure on the bearings.	Rape oil 50 per cent., lard oil 25 per cent., and tallow oil 25 per cent
For moderately high-speed shafts with very heavy pressure on the bearings.	Sperm oil 35 per cent., rape oil 20 per cent., lard oil 25 per cent., tallow oil 15 per cent., and fine plumbago 5 per cent.
For low-speed shafts with very heavy pressure on the bearings.	Graphite-grease, composed of 25 per cent. of tallow, 30 per cent. of palm oil, 33 per cent. of mineral oil, and 12 per cent. of fine graphite or plumbago.

The addition of graphite to lubricants reduces friction, and diminishes the cost of lubrication. Graphite, grease and other thick lubricants are only suitable for the journals of shafts having considerable play in the bearings.

Oils of inferior quality are not economical, and they sometimes contain acids which injure some metals, especially those containing much zinc and lead.

Heating of Journals and Bearings.—The heating of a revolving journal is produced by the friction of its surface on that of its bearing.

The intensity of the pressure on a bearing surface decreases as the length of a journal increases; and the heating tendency decreases as the area carrying off the heat is increased.

Steam Engine Erection. One of the main points to be observed in designing an engine foundation is, to have the bottom of sufficient dimensions to prevent vibrations, or settlement. The functions of an engine foundation are three fold, viz:

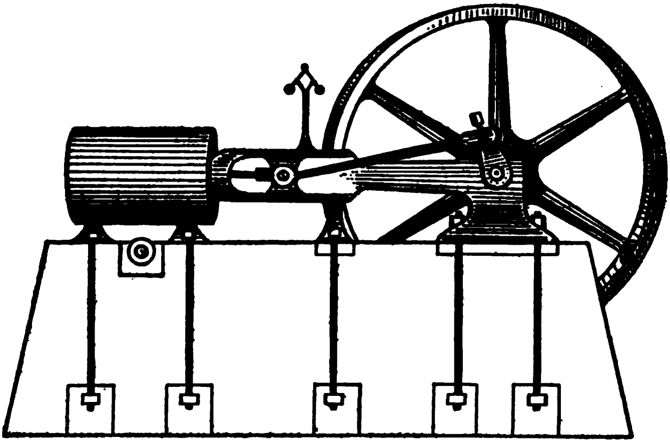


FIG. 112.

I.—To support the mass of the engine at a given height or level.

II.—To preserve alignment of engine with the main shaft.

III.—To absorb vibrations of the moving parts of the engine.

Construction of Engine Foundations.—The foundation is to absorb the vibrations of the engine. It is therefore necessary that it be constructed in a single mass, or at least be so connected that one portion cannot move independent of the other portions. For this reason it is best to build the foundation of concrete in a single block if possible. A good, well built concrete foundation is preferable to any other kind, although brick or stone may be used to advantage if these materials are cheapest. Where the importance of the structure warrants it, small tunnels, or thoroughfares, should be made or left through the mass of the foundation by means of which access may be had to the lower ends of the bolts by which the bed-plate is bolted to the masonry (see Fig. 112).

For small engines, footings of rough masonry or ashlar may be used to distribute the pressure, and on this footing the foundation proper of brick may be built. The third plan is to make the foundation a monolith of concrete. Upon a proper footing to distribute the weight, a box of rough boards without top or bottom is laid, and within it successive layers of cement concrete are thrown in and well rammed until the desired height is reached. When brick is used it should be of first quality, hand-burned, and laid in cement-mortar. Common lime-mortar is liable to crumble and disintegrate under vibration, and the whole principle of the foundation is to have it act as a solid mass. When appearance is to be considered, the face of the brick foundation may be made of face, or Philadelphia brick, while the interior is of ordinary grades. Since the bed-plate is to be bolted to the foundation, the greatest care must be observed in locating the necessary bolts in their proper places.

Footings to Prevent Vibration.—The mass which it is convenient to get in a vertical engine bed-plate is often not enough to provide for the absorption of all vibration. The vertical engine, when chosen because floor-space is to be saved, does not call for extended area in the foundation, so that sufficient mass can only be gotten by going deep. Where this is inconvenient, or where rock is struck, engineers have had to provide special footings to arrest vibration. It has been tried by some to underlay the foundation proper with timber or rubber, but a springing material of a class to which these belong often causes the very difficulties they are designed to prevent.

Vibration of machinery, or any solid substance is of two sorts: the material either swings crosswise as in the vibrating string of a musical instrument or in a flapping belt, or the motion of the particles is lengthwise or parallel to their long axis. If the oscillation, or vibrating period of the material used as an absorber of vibration happens to coincide with the vibration period caused in the engine frame by the speed of reciprocation, or by the belt flap, the deadener partakes, and multiplies the objectionable vibration. Probably no better material is to be found for the purpose of stopping vibration than sand, provided it can be kept dry, and all motion prevented, and the foundation-block itself is of sufficient mass. The foundation pit should be dug two or three feet deeper, and two or three feet wider than the foundation proper is to be. This pit is surrounded with the proper sheathing to prevent the displacement of the sand, which is filled in two or three feet below the bottom of the foundation, and then around it on the sides as it is built up. Hair-felt or mineral wool layers have been used underneath the footing-course. But if the foundation-block is not massive enough, these methods only serve to aggravate the difficulty which they are intended to cure. Very satisfactory results have been obtained abroad from the use of asphaltic concrete for massive footings.

This material possesses a certain sort of elasticity with its massive character, and its period of vibration is so definite, and so much shorter than the period of the engine's vibrations, that the latter are broken up and neutralized before they reach the transmitting rock, or hard-pan. Most annoying vibrations are often caused in high-speed engines by the impact of steam in an exhaust-pipe with elbows. The difficulty is intensified when there are water, or oil drops in the exhaust current. Their impact against the elbow which deflects them will set lengths of pipe a-tremble, and their motion will be transmitted over a very extensive area.

Foundation Bolts.—The bed-plate of the engine requires to be strongly and rigidly secured to the foundation in order that the latter may act with the bed-plate as one mass, and thus prevent the latter from moving on the foundation.

These anchor bolts vary in size with the size of the engine, but they should never be of such small diameter that it can be possible to twist them off with an ordinary wrench. Common diameters of anchor bolts for engines of medium size are from 1⅛ to 1½ inches.

The larger sizes of engines require 2-inch bolts, while for the smallest sizes, ¾-inch bolts are sufficiently strong, if properly handled in screwing down the nuts. Regarding the lengths of anchor bolts, it is best to have them extend all the way to the bottom of the foundation, if possible, in order that the upward strain upon them may be widely distributed in the foundation.

The location of these bolts in the foundation is determined by the holes in the bed-plate through which they are to pass. The typical bed-plate as a general rule has bolt holes at the cylinder end, in the feet, and at the crank shaft (see Fig. 112). The bolts, furthermore, must be built into the foundation, and be of such length that when the foundation is completed, and the bed-plate placed upon it, the upper ends of the bolts will protrude through the holes in the bed-plate a sufficient distance to receive the nuts which these upper ends are to carry.

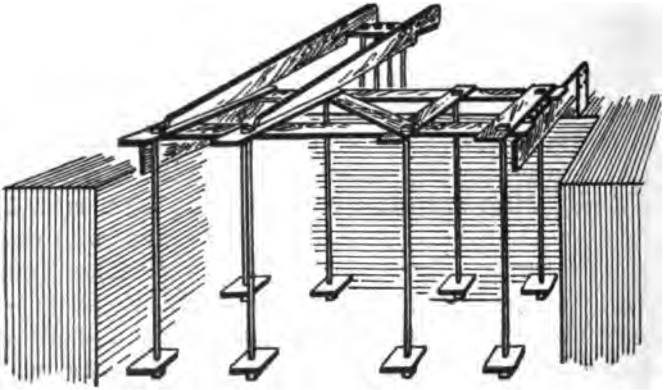


FIG. 113.

The method used to accomplish this is illustrated in Fig. 113, which presents a typical arrangement for this purpose. The wooden frame is called a template. It has holes made through it at points which correspond to the holes in the bed-plate, and when the nuts are in place on the upper end of the bolts the template is adjusted to the proper height above the datum plane, or plane of reference, and the foundation is built around the hanging bolts. The lower ends of the bolts are fitted with thread and nuts, on top of which rest the bearing or distributing plates or washers of cast or wrought iron. The distributing-plate is to enable the effort of the bolt to be borne by a number of bricks without danger of pulling through, and the nut and thread permit a vertical adjustment of the bearing-plate so that it shall come at the under surface of a joint in the coursed masonry. To permit of a certain limited horizontal adjustment of these foundation-bolts, each bolt should be surrounded with a length of pipe reaching from the bearing-plate to the top of the masonry. The diameter of this pipe is so chosen that the bolt can be deflected within the hole which the pipe makes, and after the bolt is in place and the alignment completed, the space between the bolt and the sides of the pipe is filled with concrete, and the position of the bolt is fixed. The template in Fig. 113 shows the anchor bolts for the outboard pillow-block attached to the principal template. This is usual when drawings of the template are furnished by the engine builder, and it is desired to make the foundation all in one block.

In cases where the length of the engine shaft makes it desirable to have a separate foundation for the outer bearing, it is usually more convenient to work with an independent template.

Constructing a template.—A template is usually made of 7/8-inch boards fastened together with clinch nails or screws. Fig. 114 shows a template layout as arranged for an engine of the detached outboard pillow-block type.

In laying out a template, just locate the center lines of the engine, and crank shaft upon a smooth platform

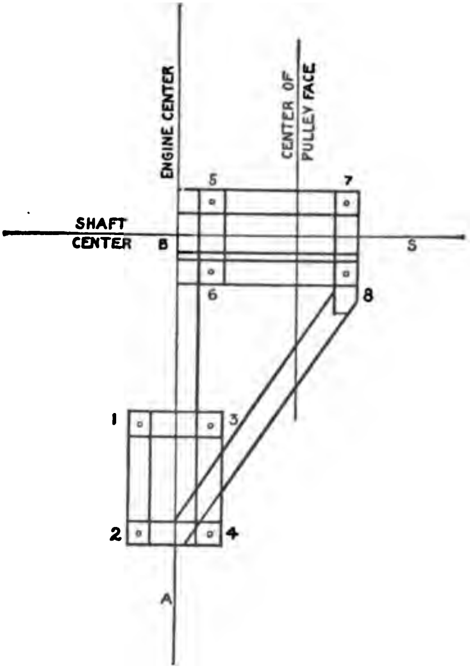


FIG. 114.

or floor sufficiently large to permit full dimensions of the engine bed plate and pillow block including the anchor bolt holes to be measured upon it. Referring to Fig. 114, first draw center lines A, B, and B S on the floor, the one representing the center line of the engine, the other that of the crank shaft. The proper locations of the holes for the anchor bolts, 1, 2, 3, &c., are obtained by measurements from lines A, B and B, S, corresponding to measurements made on the bed plate itself. These distances are laid out on the floor or platform in exact accordance with those on the bed plate, and it then becomes an easy matter to locate the centers of the holes on the template. The construction of templates will, of course, vary according to the particular type of engine to be erected, but a general idea of the methods to be pursued in the work of building and properly locating the template, and also of building the foundation around the anchor bolts, may be obtained by a study of Figs. 113 and 114. One thing should always be borne in mind regarding this work, viz., that measurements and the locating of centers must be exact in every detail. Otherwise trouble will be encountered when placing the bed plate upon its foundation. Of course if the anchor bolts are allowed a certain amount of lee-way in the foundation, as has been already described, they may be deflected to some extent to allow of their passing up through the holes in the bed plate.

Setting up a Template. The best method of setting up a template is to suspend it from over head, although the one illustrated in Fig. 113 is shown as supported from underneath. The advantages of overhead suspension are, that all the space around the foundation is left free, and there is nothing to interfere with the men in their work; also there is practically no danger of the template becoming displaced from its proper location when suspended from overhead, while with the underneath support there is danger that through accident, or carelessness on the part of the workmen, the template may be slightly moved from its true location, and this would cause trouble.

After the foundation is finished and the top is leveled accurately, the template, upon which the center lines A, B and B, S are represented by the straight edges of the two boards as shown in Fig. 114, may be placed on the top of the foundation, and other measurements for shafting and other purposes taken from it. Care should be exercised in the selection of lumber for use in constructing a template. The lumber may be white pine, yellow pine or spruce, and it should be perfectly dry. If no dry lumber is available, it should be dried in some way before using, for the reason that if green lumber is used there is danger that it will shrink and warp, and this will destroy the accuracy of the template. Another important point to be remembered in constructing a template is, to be sure that it is braced sufficiently well to keep it in exact shape as laid out in the design.

Alignment of the Template. The foundation, or anchor bolts bear a certain relation to the axis of the cylinder. The axis, or center line of the cylinder must be in a plane truly at right angles with the center line of the crank shaft. If the engine shaft is to drive a line shaft by belting, or gearing, these two shafts must be exactly parallel. Hence it is of prime importance to have the cylinder axis perpendicular to the line of shafting, and the template which carries the anchor bolts must be very carefully placed or oriented with respect to these determining lines.

The drawing furnished by the builder of the engine from which the template is to be made usually has on it the center-line of the cylinder, so that it can be laid out upon the boards of the template.

For the obtaining of the vertical plane through the cylinder-axis a line stretched over the foundation-pit and carried to suspended plumb-bobs is the usual device. For laying off the center-line of shafting or wall-lines the expedient of snapping a chalk-line upon the floor is the most convenient. The centers of the shaft are transferred to the floor by plumb-lines, or offsets may be taken from permanent walls. Such center-lines having been established, the plane at right angles to it is established by points and lines using either a transit with graduated horizontal limb and making repeated readings, or by the ordinary geometric methods, or by the use of a massive T square whose head and blade exceed six feet in length and whose squareness has been carefully verified.

If a pulley or fly-wheel has been placed upon the line-shaft to which it is desired to draw a perpendicular line, a most convenient method is to stretch a twine or fine wire across the diameter of the pulley as nearly as the shaft will permit. With pulleys which have been turned, the edges of the face determine a plane perpendicular to the axis, so that the tense string touching the face at one point will only touch the face at a point on the other side of the shaft when the further end of the string lies in a plane which is perpendicular to the axis. This same method is a very convenient one to extend for the purpose of bringing two shafts parallel to each other where both carry pulleys.

Locating the Bed-plate on the Foundation.—The foundation being completed, the bed-plate is to be lifted upon it and dropped into place with the bolts passing up through the holes in the bed-plate. Where cranes or similar lifting appliances are a feature of the power-house equipment this process becomes simple. In their absence the bed-plate must be lifted by jacks and blocking high enough to clear the bolts. It must then be rolled on skids into place by the successive withdrawing of the blocking.

The masons or bricklayers who have built the foundation do not usually have appliances for working to as close dimensions or as accurate levels as the setting of the engine requires. Furthermore, the top of the foundation is rarely a true plane, while the bottom of the bed-plate is very nearly a plane as a rule. It is necessary, therefore, to make a joint between the bed-plate and the masonry work which shall support the bed-plate all over and in a plane as nearly level as it can be made. This process is so much easier when the brick-work or jointed masonry is covered by a single flat cap-stone, that where the dimensions of the foundation permit its use is to be preferred. It is usually a sawed or planed slab of bluestone or flagstone from four to six inches thick, and a little larger than the foundation pier to which it serves as a finish or coping. The holes for the foundation bolts must be drilled in it, and it is lowered to its place upon a good bedding of cement. In the absence of such a cap or coping the bearing of the bed-plate comes upon a surface which is full of joints. The bed-plate is lowered over the foundation bolts, and rests upon thin flat shims, or wedges of metal, which are placed on each side of the bolts between the bed-plate and the foundation. The nuts of the bolts are then screwed home, compressing the shims, while the bed-plate is carefully levelled as the strain is taken at each bolt. By driving in or loosening the shims any distortion or warping of the bed-plate by the bolts is prevented, and the bolts are tightened home until they refuse to go further.

The bed-plate is now rigidly bolted to the foundation and rests upon a number of points in a plane. Between the bed-plate and the foundation is a space between the shims equal to their thickness, and this joint requires to be filled. The materials used for this purpose in setting a bed-plate and making the joint are five. They are methods applicable to the setting of any machinery.

1. Shredded oakum may be driven into the joint with a chisel, as the seams of wooden vessels formerly were calked. This makes an elastic joint, but it lacks permanency.

2. Felted hair is used in the same way and has the same properties.

3. A rust-joint, as it is called, may be used. This is made by taking a thin cement grout into which cast iron borings or chips are introduced with a little powdered sal ammoniac and flour of sulphur. A dam of putty or clay is made around the outside of the bed-plate, and this mixture run into the joint and well

worked in with a trowel. The rusting metal unites the mixture to the iron, and the cement to the stone.

4. The sulphur-joint. This is one of the most widely used methods for bedding the engine. A clay or putty dam is made around the bed-plate, and the ordinary roll sulphur melted in an old kettle and poured into the joint between the bed and the masonry. It expands on solidification somewhat like ice to fill every interstice and give full support to the bed-plate. It undergoes no deterioration from oil or vibration. If care is not taken in melting the sulphur, it will become too hot and begin to oxidize, giving off an offensive gas. Sulphur in melting becomes fluid at a comparatively low temperature, becomes more viscid as the temperature rises, and passes to a second fluidity just before it is ready to burn.

5. The type-metal joint. Advantage is taken of the property possessed by certain antimony alloys (such as Babbitt, type metal, etc.) of expanding at the moment of solidification to use them for bedding or jointing bed-plates. The method of using them is the same as that practiced with sulphur, and they are preferred by many engineers particularly for bedding the narrow feet used with Corliss bed-plates.

Alignment of Outer Pillow-Block or Shaft-Bearing.—Many engines have a bearing on each side of the crank pin and connecting rod. Such engines are called center-crank engines. The two bearings for the shaft are on the bed-plate, and the fly-wheels overhang their bearings. Side-crank engines, on the other hand, have a shaft bearing behind one crank on the bed-plate, but the outer end of the shaft requires an independent bearing. The fly-wheel or belt-wheel or both will usually be upon the length of the shaft between these two bearings.

It will be seen at once that the location of this outer bearing is of vital importance. In small engines it can be provided for approximately by the template as shown in Fig. 113, but for large engines and for its final adjustments in small engines a different method should be used.

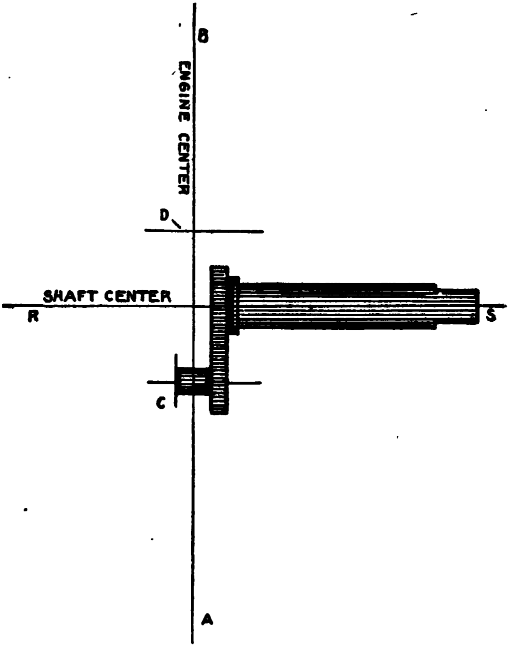


FIG. 115.

If the outer bearing is too high or too low, it will force the crank to revolve in a plane making an angle with the true vertical plane, and twist the connecting-rod in each stroke. If out of place in a horizontal plane while correctly located in a vertical plane, it will force the crank to revolve in a plane which makes an angle with the axis of the cylinder, in which case it will bend the connecting rod in each stroke; or it may be out of place in both planes, so that the connecting rod will be both twisted and bent. The effect of either or both errors of alignment of this outer bearing is to wear the crank-pin out of its cylindrical shape, and to cause a knock or pound, and heating at the joint, which no adjustment of these bearings will cure. The proper method of aligning the outer bearing involves, first, the establishment of the true axis of the cylinder after the bed-plate is in place, and the foundation-joint complete. This is best done by stretching a fine piano-wire through the empty cylinder, carefully adjusting it to the center of the bore, and fastening it tightly stretched to walls or fixed objects. To get the wire central is a matter of painstaking care and trial with gauges of wood or metal whose length is the cylinder radius. The axis of the cylinder being established, the shaft and crank are put in place in the bearing on the bed while the outer bearing is provisionally supported and located. The shaft is then turned until the crank coming towards its inner

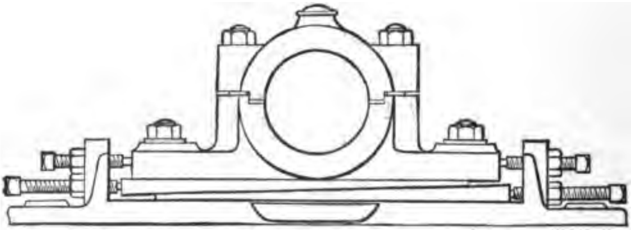


FIG. 116.

dead center touches the wire which marks the prolongation of the cylinder axis. It will touch it at a certain distance from the end of the crank pin, and from one of its collars.

Fig. 115 will serve to more clearly illustrate this method of alignment. The line drawn through the center of the cylinder is represented at A, B, while the center of the engine shaft is outlined at R, S. The crank being under the line A, B, the shaft is revolved until the crank-pin is brought as close as possible to the line without touching. While in this position the distance to the fixed collars on the pin is measured with a steel scale, or with inside calipers. The line A, B, should be equidistant between the fixed collars on the crank-pin, as shown at C. After getting this measurement, the engine shaft is again revolved until the crank-pin again comes almost to the line at D, where the distance from the line to each collar is again carefully measured, and if found exactly equal to the measurements taken at C, the shaft may be considered "square" with line A, B,

FIG. 117.

and also square with the cylinder. If the measurements at C and D do not agree, the outer bearing should be adjusted horizontally by means of the set-screws shown in Fig. 116, until the wire cuts the crank-pin at the same point, and in the same plane on both its outer and inner centers. Reference has already been made in a general way to the method of centering the line passing through the engine cylinder.

A more detailed description of the process will now be given. Engine builders usually send out with each engine, a device for use in securing the line in the center of the cylinder. This device is simply a wooden cross of the maltese type having the ends of its four arms rounded off to fit the circle of the cylinder bore, the length of each arm being just equal to the diameter of the bore. The two arms are halved together at the center as shown in Fig. 117, and a one-inch hole is bored through the center. Over this hole is tacked a piece of tin or sheet metal, having a small hole through its center just large enough to allow of passing the fine wire line A, B, Fig. 118, through it. This small hole must be the exact center of the cross, and if it be necessary for the millwright to make a cross, the hole is the center from which the circles on the ends of the arms must be inscribed. When the wooden cross is properly fitted to the cylinder bore, and is secured in the back, or head end of the bore, the cylinder head being off, and the piston and rod having been previously removed, the wire line is passed through the small hole in the center of the cross, and secured permanently to a wooden arm held by screwing a nut down on it. The other end of the line should extend some distance past the crank end of the bed plate, and there be temporarily secured in such a position as to cut both the center of the crank shaft, and a point midway between the collars of the crank pin; the locating of this end of the line in its temporary position may be accomplished by eye-sight. Having stretched the line taut the next step is to adjust the line in such a position that it will pass exactly through the center of the cylinder bore from end to end. Knowing that the line is in the exact center of the head end at B, C, Fig. 117, the second step will be to center it where it passes through the piston rod stuffing box in the crank end cylinder head. The method of doing this is illustrated

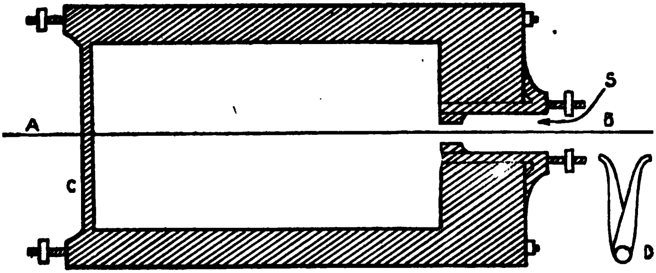


FIG. 118.

in Fig. 118. The line A, B, is shown passing through the wooden cross C, and on through the stuffing box, which being accurately bored out, will serve as a guide for adjusting the line by means of the inside calipers D shown in the cut, and applied at or near point S. Any adjustments of the line must of course be made at the outer, or B end, the A end being already centered by means of the wooden cross.

Having finally succeeded in centering the line through the engine cylinder, the third step is to bring the center of the crank shaft, and the crank pin to conform with the line as previously described.

Care should be taken, both in setting up the engine, and also in adjusting the main bearings, to see that the engine shaft is square, as above described, and that it remains so.

Where adjustment is required the usual procedure is followed of correcting half the error and testing the alignment again. The outer pillow-block is often made to rest upon a special foundation-plate which has provision for the adjustment upon it of the bearing proper in the horiznotal plane (Fig. 116). The vertical adjustment otherwise is made either by shimming or filling in with sulphur or type metal below the plate, and the last and finest adjustment can be made by liners underneath the bearing-brasses. The alignment of vertical engines is usually simpler than that of horizontal engines, because the bearings are always on the bed-plate and have been made right as to alignment by the builders in their shop-handling. The alignment in the erection of beam-engines is a simple and obvious extension of the prinples laid down above. The vertical cylinder-axis, and the vertical line through the center of the crank pin when the latter is at the top, and at the bottom of its travel, determine the vertical plane in which the beam must play, and the crank pin at its 90 degrees and 270 degrees point must remain in that same plane.

The adjustment of the horizontal plane may be effected by a sensitive level if the engine has also been leveled in the plane at right angles to the cylinder-axis in setting upon the foundation. A more sensitive and satisfactory vertical adjustment of the out-board bearing may be accomplished by placing the crank pin at 90 degrees from its dead center, and holding a plumb line so as to touch the wire at the pin, then noting the distance of the vertical plane thus established, from the end of the pin, or collar. If the plumb line touching the wire also touches the crank pin at the same distance from the reference mark when the pin is at half stroke below the wire, then the pin is turning in a vertical plane through the wire, and the outer bearing requires no vertical adjustmnt.

Equalizing the Clearance. An important feature to be taken into account during the erection of an engine, and in fact in its operation is the equalization of the piston clearance at each end of its stroke.

This means that the distance of the piston from the cylinder head when the engine is on either dead center must be as nearly as possible the same for either end of the stroke. Strict attention should be given this matter also during the operation of the engine, as when keying up either the crosshead or crank pin bearings after they have become somewhat worn there is danger of drawing the piston too near the crank end of the cylinder. This may be obviated however by the insertion of thin liners next to the brasses.

However the point that most concerns the millwright, or the erecting engineer, and for which he will be held responsible, is the equalization of the clearance before the engine is started to running.

After the work of alignment is completed, the piston should be placed in the cylinder, and the piston rod keyed into the crosshead, after the latter has been accurately centered in the guides. The adjustment of the crosshead is a matter which should be given strict attention also, in order that the center of the hole in the boss for the reception of the piston rod may be exactly in line with the center of the cylinder. Adjustments are made by means of set screws and adjustable shoes with which all large engines should be provided. A large pair of inside calipers are needed for this work. The main or connecting rod should then be placed in position, and properly keyed in its bearings, and the cylinder head put in its place and secured there temporarily by tightening up several of the nuts at opposite points. The engine is now ready for equalizing the clearance, and this may be accomplished in several ways. Probably the most accurate method is by means of two pieces of lead wire or even small lead pipe inserted into the cylinder through the holes drilled for the indicator pipe nipples, or if there are no indicator openings provided, the lead wire may be inserted through the holes provided for the cylinder cocks.

Having inserted the pieces of lead far enough into the cylinder to insure the piston striking them, turn the engine first to one center, and then on to the other center, after which withdraw the lead pieces and by means of calipers, measure the thickness of each. If they have not been flattened to the same thickness, as for instance if the one that was in the crank end of the cylinder is thinner than the other it indicates that the piston approaches nearer to that head than it does to the other head, and this may be remedied by moving the crank shaft toward the cylinder a distance corresponding to one half the difference between the thicknesses of the two pieces of lead. Or the adjustment may be made by the insertion of thin liners behind the crosshead, or crank pin brasses as previously described.

BOILER SETTING.

It generally falls to the lot of the millwright to superintend the setting of the boilers, and in many cases the installation of the piping also. Therefore a space should be devoted to a discussion of these two very important topics.

Location of Boilers. The location of the power plant, especially the boilers, is one of the most important problems in connection with the designing and erection of a factory or mill.

If the factory is a wood working establishment, then the question of getting the shavings and other refuse to the boiler room by the shortest route, and in the most economical manner must be taken into account.

If coal is to be used as fuel, the problem of transporting it to the boiler room, and the removal of the ash and cinders is to be considered. Then there remain other factors to be dealt with, such as steam heat for the buildings; steam to be supplied for drying; for chemical purposes; for operating steam pumps, and other steam machinery. The problem to be solved is the generation of steam, and the transmission of the power at the lowest possible cost.

The efficiency of a power plant depends largely upon the boilers, and the methods employed in their operation. Again the efficiency of the boilers as generators of steam depends in a large measure upon their location, and erection. Heat losses should be guarded against in every way possible. Ample room should be provided above the boilers for the piping, and facilities allowed for repairs to the same, and for washing out, and cleaning the boilers.

Setting. In the following remarks concerning boiler setting, reference is had chiefly to the horizontal tubular boiler. Owing to the many and varied styles of water-tube boilers no prescribed set of rules is applicable, each builder of this type of boiler having a set plan of his own for the brick work. In the case of internally fired boilers the matter of setting resolves itself into the simple point of securing a sufficiently solid foundation, either of stone or brick laid in cement, for the boiler to rest upon. Concrete makes the best foundation for steam boilers especially in cases where the nature of the soil is spongy or soft. In the case of installations where a number of boilers are to be set in battery and the columns of the building are placed among the boiler foundations, it is necessary to place heavy and absolutely safe foundations for the boiler walls in order to prevent settling. Regular setting plans should be drawn, and followed in setting boilers. The boiler setting should in fact be laid out and squared with the building line, in the same manner as with the other machinery.

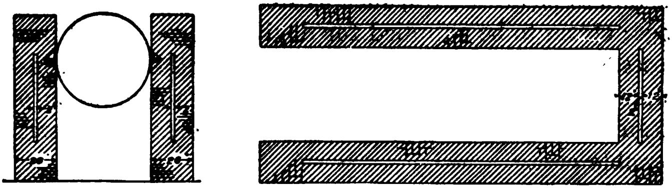


FIG. 119.

Having laid out and staked off the ground on which the boilers are to stand, the next step is to roll the boilers in place, jack them up a sufficient height, and build under and near each end a supporting crib-work of blocks 3 or 4 feet long by 5 or 6 inches square. Care should be taken in placing the boilers upon their temporary foundations of blocks, to see that they are exactly in the places they are to occupy permanently, that is, so far as vertical and horizontal distances are concerned.

Horizontal tubular boilers should always be set at least one inch lower at the back end than at the front, to make sure that the rear ends of the tubes will be covered with water so long as any appears in the gauge glass, provided of course that the lower end of the glass is properly located with reference to the top row of tubes.

Usually it is not necessary to excavate very deep for a boiler foundation. A trench a foot or so in depth, following the outline of the walls, and across where the bridge wall is to go, will be sufficient in most cases. The side walls, including the air space, are about 30 inches, and the rear wall, 28 inches. The bridge wall is abcut the same thickness, and the front wall may be 9 inches for a half arch front, or 16 to 18 inches for a full flush front boiler.

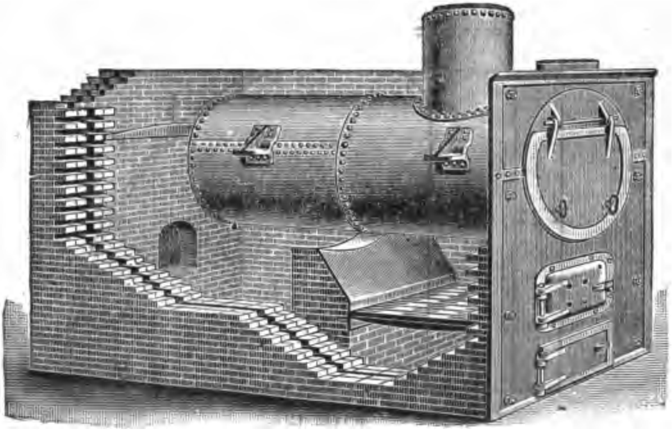


FIG. 120.

RETURN TUBULAR BOILER—SHOWING SETTING.

Buck Stays and Tie Rods. After the concrete, or stone foundation has been laid, and before the brick work is started, the buck stays and lower tie rods should be located, and the latter placed in position. Boiler walls should always be well secured in both directions by tie rods extending throughout the entire length and breadth of the setting, whether there be one boiler or a battery of several. The bottom rods should be laid in place at the floor level when starting the brick work, and the top rods extending transversely across the boilers can be laid on top of the boilers. The top rods extending from front to back can be laid in the side walls or rest on top of them. All tie rods should be at least one inch in diameter, and for batteries of several boilers they should be larger. The rods should extend three or four inches beyond the brick work, with good threads and nuts on each end to receive the buck stays. In laying down the transverse tie rods they should be located so as to allow the buck stays to bind the brick work where the greatest concentration of heat occurs.

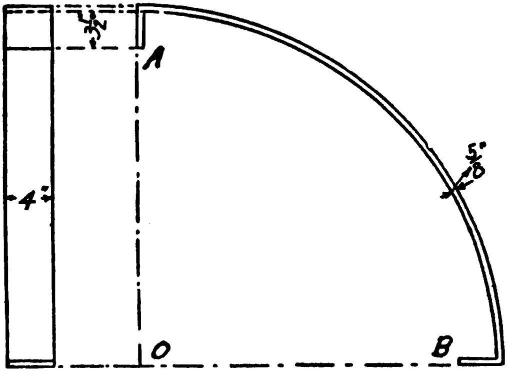


FIG. 121.

In the case of the horizontal tubular boiler, there are two methods of support, one by suspension from I-beams and girders, to be described later on; the other by supporting the boiler upon brackets or lugs, riveted to the side sheets, and resting upon the side walls, and in such settings particular attention should be paid to securing a good foundation for the walls, and great care exercised in building them in such manner that the expansion of the inner wall or lining will not seriously affect the outer walls. This can be done by leaving an air space of two inches in the rear and side walls, beginning at or near the level of the grate bars and extending as high as the fire line, or about the center line of the boiler. Above this height the wall should be solid. Fig. 119 shows a plan and an end elevation illustrating this idea. The ends of some of the bricks should be allowed to project at intervals from the outer walls across the air space, so as to come in touch with the inner walls.

Where boilers are set in batteries of two or more the middle or party walls should be built up solid from the foundation. All parts of the walls with which the fire comes in contact should be lined with fire brick, every fifth course being a header to tie the lining to the main wall.

Bridge Walls. The boiler setting shown in Fig. 120 shows the bridge wall as being curved on top to conform to the circle of the boiler shell. Concerning this problem, much difference of opinion exists among engineers. The author's experience of many years has confirmed him in the belief that bridge walls should be built straight across from wall to wall of the setting, and should not be curved to conform to the circle of the boiler shell. The proper distance from the top of the bridge wall to the bottom of the boiler varies from eight to ten inches, depending upon the size of the boiler. The space back of the bridge wall, called the combustion chamber, can be filled in with earth or sand, and should slope gradually downward from the back of the bridge wall to the floor level at the rear wall, and should be paved with hard burned brick. The ashes and soot can then be easily cleaned out by means of a long-handled hoe or scraper inserted through the cleaning out door, which should always be placed in the back wall of every boiler setting.

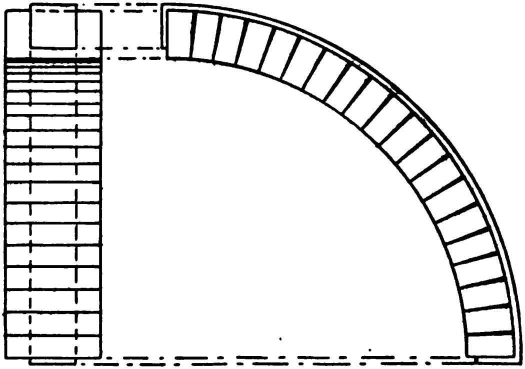


FIG. 122.

Bearings for Lugs.—Upon the brick work and immediately under each lug of the boiler there should be laid in mortar a wrought or cast iron plate several inches larger in dimension than the bearing surface of the lug and not less than one inch in thickness. Upon each of these plates there should be placed two rollers made of round iron 1 or 1⅛ in. in diameter, and as long as the width of the lug. These rollers should be placed at right angles to the length of the boiler, in such a position that the lug will bear equally upon them. The object of the rollers is to prevent disturbance of the brick work by the endwise expansion and contraction of the boiler.

Grate Surface.—The number of square feet of grate surface required depends upon the size of the boiler. A good rule and one easy to remember is to make the length

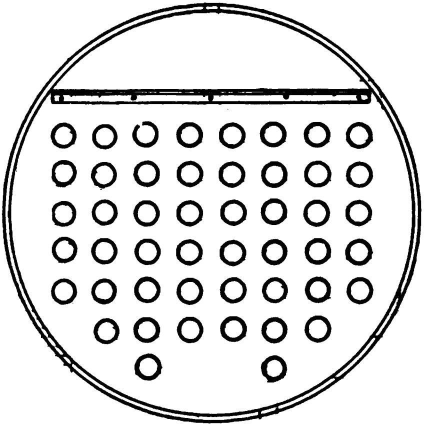


FIG. 123.

of the grates equal to the diameter of the boiler. The width, of course, will depend upon the construction of the furnace. If the fire brick lining is built perpendicular, the width of grate will be about equal to the diameter of the boiler. On the other hand, if the lining is giving a batter of three inches, starting at the level of the grate, then the width will be reduced six inches. It is customary to allow one square foot of grate surface to every 36 sq. ft. of heating surface. The distance of the grate-bars from the shell of the boiler varies from 24 to 28 in., according to the dimensions of the boiler.

Blow Off Pipes.—Blow off pipes should always be connected with the lowest part of the water space of a boiler. If there is a mud drum, then of course the blow off should be connected with it; but if there is no mud drum, the blow off should connect with the bottom of the shell, near the back head, extend downwards to the floor of the combustion chamber, and thence horizontally out through the back wall, where the blow off cock can be located.

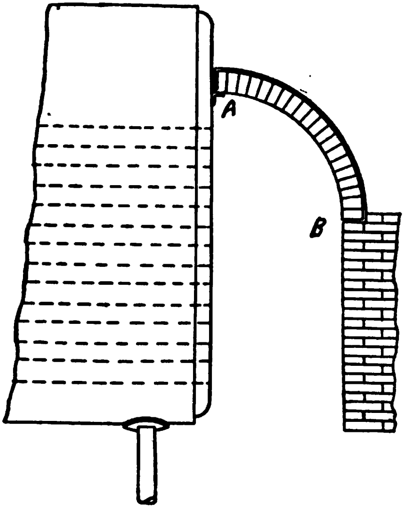


FIG. 124.

Back Arches.—A good and durable arch can be made for the back connection, extending from the back wall to the boiler head, by taking flat bars of iron ⅝ x 4 inches, cutting them to the proper length and bending them in the shape of an arch, turning 4 inches of each end back at right angles, as shown in Fig. 121. The distance O-B should equal that from the rear wall to the boiler head, and the height, O-A, should be about equal to O-B, and should bring the point A about two inches above the top row of tubes. The clamp thus formed is filled with a course of side arch fire brick, Fig. 122, and will form a complete and self-sustaining arch, the bottom, B, resting on the back wall, and the top, A, supported by an angle iron riveted across the boiler head about three inches above the top row of tubes. See Figs. 123 and 124.

Enough of these arches should be made so that when laid side by side they will cover the distance from one side wall to the other across the rear end of the boiler. A fifty-four-inch boiler would thus require six clamps, a sixty-inch boiler seven clamps, and a seventy-two-inch boiler would require eight clamps; the length of a fire brick being about nine inches. In case of needed repairs to the back end of the boiler the sections can be lifted off, thus giving free access to all parts, and when the repairs are completed the arches can be reset with very little trouble and much less expense than the building of a solid arch would necessitate. This form of segmental arch allows ample freedom for expansion of the boiler, in the direction of its length, without leaving an opening when the boiler contracts. It will be seen that within all possible range of boiler movement in either direction the arch will, with this construction, always remain close to the head.

The crosswise construction of arch bars, while affording equal facility in repair work, is necessarily more expensive than the form here described, and is also open to the objection that it cannot follow the contracting boiler and maintain a tight joint or connection between the back arch and the rear head above the tubes.

Suspension Method.—A great improvement is made when we suspend such boilers from I-beams supported by cast iron columns. Figures 125 and 126 show the

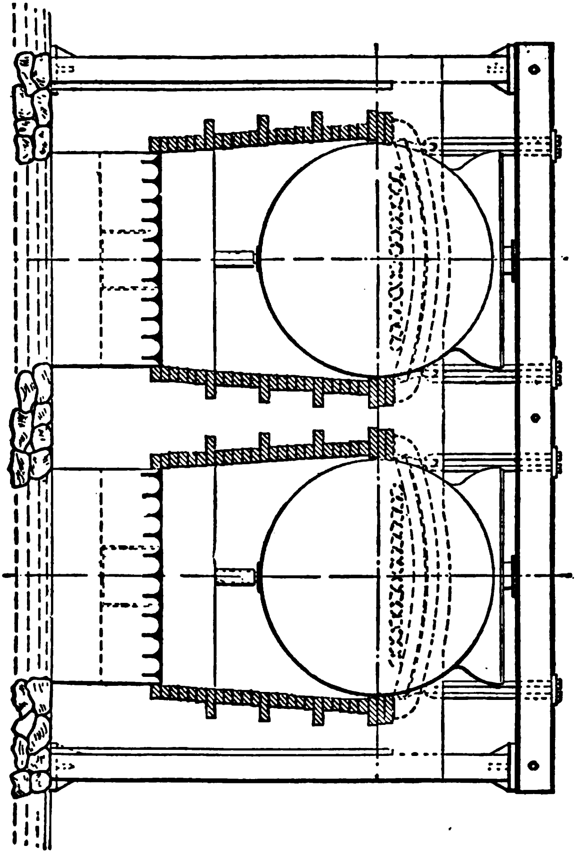


FIG. 125.

setting of this type of boilers either singly or in double batteries, by means of suspension. In setting an even number of boilers, as six or eight in one setting, it is best to divide them into pairs so that not more than two boilers will be suspended between supports.

The principal reason for this is that when the large sizes, such as from 150 to 250 horsepower, are used, the size I-beam required to safely carry this load between supports is so large that it overbalances the cost of two or more cast-iron columns.

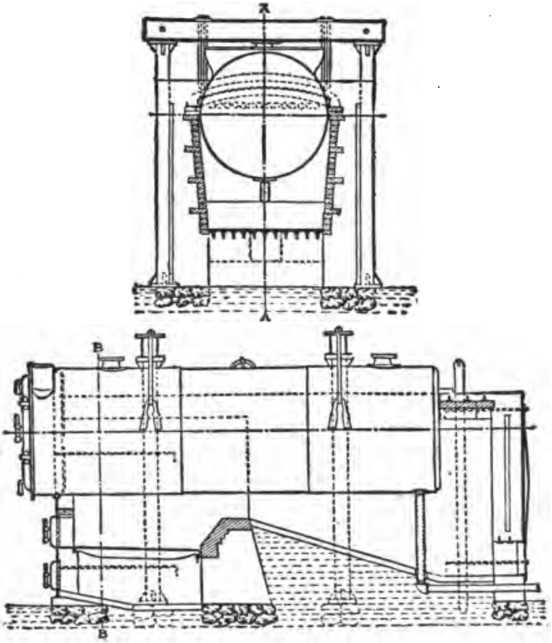


FIG. 126.

In setting an odd number of boilers, such as three or five, in a battery, columns are usually placed between the boilers with a 2-inch air space all around the column, and an air duct at the bottom of the setting which runs through from the front to the back and connects with each air space around the column. This allows a free circulation of air, thus tending to keep the columns comparatively cool. In setting boilers in this way the columns and I-beams are set in position first. The boiler is then hoisted to the proper height by means of tackle, which is attached to the I-beams, and when the boiler is brought to the proper height, the U-bolts are slipped into place and fastened by nuts and washers to the I-beams.

This method abolishes the use of blocking and leaves all of the space under the boilers clear for the brick work.

The expansion is easily taken care of by the U-bolts and hangers, and if the walls are properly set, they will show no cracks as they carry no weight, and are free to go and come. The accompanying table, No. 35, has been carefully worked out with a factor of safety of 5, and gives the different sizes and lengths of I-beams and columns required for boilers of 36 inches in diameter and 8 feet long, to boilers of 90 inches in diameter and 20 feet in length, giving the total weight to be supported and the sizes, weights, and positions of the columns, and I-beams required:

TABLE 35.

SIZES AND WEIGHTS OF COLUMNS AND I-BEAMS REQUIRED IN SETTING RETURN TUBULAR BOILERS.

HORSE POWER	15	20	25	30	35	40	45	50	60
Dia. of boiler in inches	36	36	42	42	44	48	50	54	54
Length of tubes in feet	8	10	10	12	12	12	13	13	15
Length of curtain sheet in inches	11	11	12	12	12	14	14	14	14
Total weight of boiler and water	6500	7500	9400	10500	11500	13300	14200	15800	17800
Rear head to center of hanger	2-0	2-6	2-6	3-0	3-0	3-0	3-3	3-3	3-0
Center to cen. of hang'rs	4-0	5-0	5-0	6-0	6-0	6-0	6-6	6-6	7-6
Front head to center of hanger	2-0	2-6	2-6	3-0	3-0	3-0	3-3	3-3	3-0
Distance between C of supports (1 boiler)	6-6	6-6	7-0	7-0	7-2	7-6	7-8	8-0	8-0
Distance between C of supports (2 boilers)	11-8	11-8	12-8	12-8	13-0	13-8	14-0	14-0	14-8
Length of I-beam for 1 boiler	7-3	7-3	7-10	7-10	8-0	8-4	8-6	8-10	8-10
Length of I-beam for 2 boilers	12-6	12-6	13-8	13-8	14-0	14-8	15-0	15-10	15-10
Size of I-beam required for 1 boiler	4	4	5	5	5	6	6	6	6
Size of I-beam required for 2 boilers	6	6	8	8	9	9	9	10	10
Weight per ft of I-beam for 1 boiler	7.5	7.5	9.75	9.75	9.75	12.25	12.25	12.25	12.25
Weight per ft. of I-beam for 2 boilers	12.25	12.25	18	18	21	21	21	25	25
Length of cast-iron col.	8-0	8-0	8-6	8-6	8-8	9-3	9-5	10-0	10-0
Outside dia. of C. I. col. for 1 boiler	4	4	4	4	4	5	5	5	5
Outside dia. of C. I. col for 2 boilers	5	5	5	5	5	6	6	6	6
Size of flange on ends of col. for 1 boiler	9½	9½	10	10	10	10½	10½	10½	10½
Size of flange on ends of col. for 2 boilers	10½	10½	12	12	12½	12½	12½	13½	13½
Thickness of C. I. col. for 1 boiler	½	½	½	½	½	½	½	½	½
Thickness of C. I. col for 2 boilers	⅝	⅝	⅝	⅝	⅝	⅝	⅝	⅝	⅝

HORSE POWER	70	75	80	90	100	125	150	175	200	200	225	225	250
Dia. of boiler in inches	60	60	60	66	66	72	72	78	78	84	84	90	90
Length of tubes in feet	14	15	16	15	16	16	18	18	20	18	20	18	20
Length of curtain sheet in inches	16	16	16	17	17	18	18	18	18	20	20	22	22
Total weight of boiler and water	20800	24800	27200	30300	35000	40000	44000	48000	56000	55000	67000	65000	75000
Rear head to center of hanger	3-6	3-9	4-0	3-9	4-0	4-0	4-6	4-6	5-0	4-6	5-0	4-6	5-0
Center to cen. of hang'rs	7-0	7-6	8-0	7-6	8-0	8-0	9-0	9-0	10-0	9-0	10-0	9-0	10-0
Front head to center of hanger	3-6	3-9	4-0	3-9	4-0	4-0	4-6	4-6	5-0	4-6	5-0	4-6	5-0
Distance between C of supports (1 boiler)	9-0	9-0	9-0	9-6	9-6	10-0	10-0	10-6	10-6	11-0	11-0	11-6	11-6
Distance between C of supports (2 boilers)	16-2	16-2	16-2	17-2	17-2	18-2	18-2	19-2	19-2	20-2	20-2	21-2	21-2
Length of I-beam for 1 boiler	10-0	10-0	10-0	10-6	10-6	11-0	11-0	11-7	11-7	12-0	12-0	12-6	12-8
Length of I-beam for 2 boilers	17-4	17-4	17-4	18-4	18-4	19-5	19-5	20-6	20-6	21-6	21-6	22-6	22-6
Size of I-beam required for 1 boiler	7	7	7	7	7	8	8	9	9	9	9	9	10
Size of I-beam required for 2 boilers	12	12	12	12	12	15	15	15	15	15	15	15	15
Weight per ft of I-beam for 1 boiler	15	15	15	15	15	18	18	21	21	21	21	21	25
Weight per ft. of I-beam for 2 boilers	31.5	31.5	31.5	40	40	42	42	60	60	60	80	80	80
Length of cast-iron col.	10-8	10-8	10-8	11-2	11-2	12-0	12-0	12-6	12-6	13-0	13-0	13-10	13-10
Outside dia. of C. I. col. for 1 boiler	5	5	5	6	6	6	6	6	6	6	6	6	6
Outside dia. of C. I. col for 2 boilers	6	6	6	6	6	8	8	8	8	8	8	8	8
Size of flange on ends of col. for 1 boiler	11½	11½	11½	11½	11½	12	12	12½	12½	12½	12½	12½	13½
Size of flange on ends of col. for 2 boilers	14	14	14	14½	14½	15	15	16	16	16	17	17	17
Thickness of C. I. col. for 1 boiler	⅝	⅝	⅝	⅝	⅝	¾	¾	1	1	1	1	1	1
Thickness of C. I. col for 2 boilers	¾	¾	¾	1	1	⅞	⅞	⅞	⅞	1	1	1	

Insulation.—All boilers should be well protected from the cooling influence of outside air, if economy of fuel is any object. The tops of horizontal boilers should be covered with some kind of heat insulating material, or arched over with common brick, leaving a space of two inches between the boiler and the arch. The resulting saving in fuel will far more than compensate for the extra expense in a very short time. All cracks in the side and rear walls should be carefully pointed up with mortar or fire clay. One source of heat loss in return flue boilers is short circuiting from the furnace to the breeching, caused by the arches over the fire doors becoming loose and shaky, and allowing considerable of the heat to escape directly to the stack instead of passing under the boiler and through the tubes. Another bad air leak often occurs at the back connection when the arch rests wholly upon iron bars imbedded in the side walls. This leak, as has already been noted, is caused by the expansion of the boiler, which gradually pushes the arch away from the back head until, in the course of time, there will be a space of 5/8 inch and sometimes 3/4 inch between the head and the arch. The obvious remedy for this is an arch that will go and come with the movement of the boiler, and such an arch can be secured by building it in sections, as illustrated by Figs. 122 and 124.

Coal and Ash Handling.—The question of handling fuel and ashes should always be considered when locating the steam plant. No matter how well placed the plant may be, there should be provided adequate means for handling coal and for removing the ashes. Various conditions require vastly different methods, but there are very few plants where the conveyor-elevator cannot be profitably installed for bringing in coal and carrying out ashes, the same apparatus serving both operations, and permitting the coal storage and ash dump to be located at will, either on the ground adjacent to, or removed from the building, or overhead in, or outside the boiler house.

The first cost of an installation of this kind is comparatively low, and once installed it will run for years with little cost for repairs or renewal. Therefore, apparatus of this kind should always be considered in connection with the installation of a battery of boilers.

Smoke Stacks. The smallest opening of the chimney should be one-fourth of the grate surface, and its height about 25 times this smallest opening, but never under 50 feet.

Chimneys which are built without means for ascending on the outside must have an opening at the top of not less than 24 inches. The lower diameter of a brick chimney should be 0.02 of the height of the chimney larger than that of the top opening.

Round chimneys are best because they offer less resistance to the wind, do not retard the revolving action of the gases, and having the smallest circumference for a given diameter, suffer the least from loss of heat. The bottom of the opening of the chimney should be at least 24 inches to 30 inches from the bottom of the bridging and if more than one bridging enters the chimney they should be so arranged as to give the gases from all of them the same direction, as otherwise a considerable loss of velocity would occur.

Brick chimneys are best, but also the most expensive. They last long and need no painting, nor much repairs, except pointing up once in a long while. They retain the heat much better than iron smoke-stacks.

Iron smoke-stacks are made either self-sustaining or are held by guy ropes. In the first case, the stack is provided with a strong cast-iron base, bolted down upon a foundation heavy enough to prevent the strongest wind overturning the stack. The stack is also made of heavier iron. The necessity of keeping in paint, and the danger of rusting on the inside if not continually in use, are two disagreeable conditions connected with the use of iron chimneys.

In many places iron stacks are preferred to brick chimneys. Iron stacks require to be kept well painted to prevent rust, and generally, where not bolted down, they need to be braced by rods or wires to surrounding objects. With four such braces attached to an angle iron ring at 2/3 the height of stack, and spreading laterally at least an equal distance, each brace should have an area in square inches equal to 1-1000 the exposed area of stack (diam. × height) in feet.

Stability, or power to withstand the overturning force of the highest winds, requires a proportionate relation between the weight, height, breadth of base, and exposed area of the chimney. This relation is expressed in the equation

$$C\frac{d\ h^2}{b} = W,$$

in which $d =$ the average breadth of the shaft; $h =$ its height; $b =$ the breadth of base,—all in feet; $W =$ weight of chimney in lbs., and $C =$ a co-efficient of wind pressure per square foot of area. This varies with the cross-section of the chimney, and = 56 for a square, 35 for an octagon, and 28 for a round chimney. Thus a square chimney of average breadth of 8 feet, 10 feet wide at base and 100 feet high, would require to weigh $56 \times 8 \times 100 \times 10 = 448{,}000$ lbs. to withstand any gale likely to be experienced. Brickwork weighs from 100 to 130 lbs. per cubic foot, hence such a chimney must average 13 inches thick to be safe. A round stack could weigh half as much, or have less base.

When two or more boilers are set in battery, there should be a separate damper to each boiler, and another damper in the stack or main smoke flue, and this damper should be connected to a good damper regulator which is actuated by steam or water pressure, and which is controlled by the steam pressure in the boilers.

STEAM HEADERS AND CONNECTIONS.

The design, size and location of the main steam header, and the connections between it and the boilers is a very important problem, and should receive close attention.

If it was merely a case of uniting all boiler outlets into a common pipe or header, regardless of the strains of expansion, and ease of pipe fitting, the difficulties would be few. The header should not only be a main for uniting all the boilers, but it should also be of sufficient capacity to act as a receiver-reservoir to counteract the effects of any momentarily heavy demand for steam, which would otherwise tend to lift some of the water out of the boilers. The size of the header may be determined by the following rule, viz., let sectional area of main header equal, or slightly exceed the sum of the areas of all boiler outlets connected with it. Take for example a battery of four 72 inch by 18 feet horizontal tubular boilers, each having a 6 inch outlet to be connected with main header. The area of a circle 6 inches in diameter is 28.274 square inches. Therefore the combined areas of the four outlets equals 28.274 × 4 = 113 square inches, and the area of a 12 inch pipe is 113.097 square inches. Therefore a

12 inch header will be required. It is best to have the header of a uniform diameter throughout its length, as it will then have the greatest storage capacity possible, and the supply to the largest engine, or steam user should be taken from as near the middle of the header as possible. The location of the header must be determined by local conditions, and by the positions of the boilers relative to the engine, but the header should always be so located as to give easy access to all valves, and joints in order that these may be easily manipulated from a fixed platform, or from the top of the boilers.

It should not be necessary to use a movable ladder to control any steam valve on the header, as the chances are that the ladder would not be in its place in time of emergency or accident. Locating the header in front of the boilers, over the firing space should be avoided if possible, as any leakage would be liable to discomfort the firemen. A good location is along the top of the boilers near the rear, the header being supported by brick piers built upon the boiler-setting walls.

All boiler connections should enter the header at the top and the outlets should also be taken from the top to insure that any water in the header will not be carried over to engine, but will be drained off at the proper place. It is not necessary to pitch the header for draining, provided that the drain connection is made close to the point where the heaviest draft of steam is taken. If the header is level, the movement of the steam toward the heaviest outlet will naturally cause the water in the bottom of the header to flow in the same direction. A good arrangement for draining a header is to use a cross at the heaviest outlet, with the outlet connection taken from the top opening, and the lower opening fitted with a blind flange tapped for drainage, but the use of a good high-pressure trap attached to the drain opening of the header, and discharging into a return tank, hot-well, or open heater is the most reliable method of drainage.

Valves. The question of valves is next in order. The day of the single valve in each boiler connection is passing. Many cities by ordinance now require two valves in each connection, and many engineers know only too well what it means to crawl into a boiler with a leaky valve on top of them. This condition can be eliminated by the use of two valves. Globe valves should be avoided on account of the turns in the steam path, gate, and angle valves being preferable. One of the neatest, and most

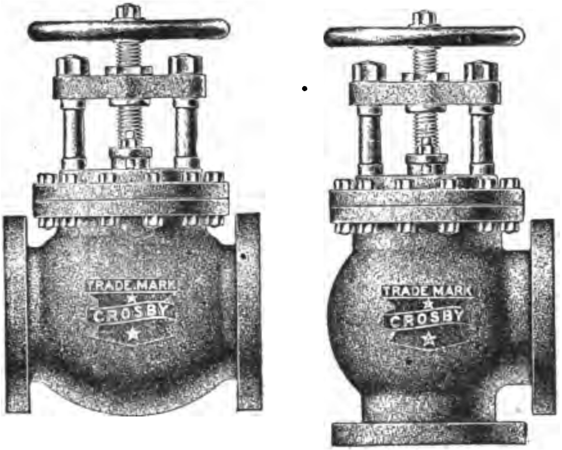


FIG. 127.

CROSBY SPRING-SEAT VALVE.

Instead of the ordinary cast iron yoke top they have steel pillars and cross bar so designed as to afford great strength and durability. Sizes, 2½ in., 3 in., 3½ in., 4 in., 4½ in., 5 in., 6 in., 7 in., 8 in., 9 in., 10 in., 12 in.

efficient arrangements is to use two angle valves, one on top of the header and the other on top of the boiler.

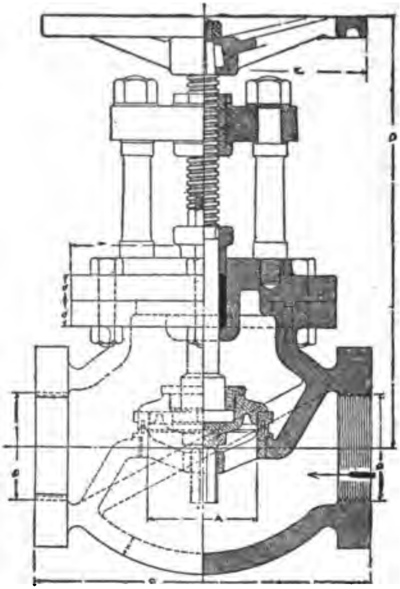


FIG. 128.

VERTICAL SECTION. SCREWED GLOBE VALVE.

UNLOADING AND SETTING UP MACHINERY.

The work of unloading machinery from car or boat and getting the machines into the building, and onto their foundations is generally performed by a gang of riggers, the millwright himself, or someone of his assistants superintending the job. In fact a millwright, in order to be successful in his chosen occupation, should be well posted in the details of handling machinery to the best advantage.

The most practical method of unloading and moving heavy machines or parts of machinery, is by means of the skid and roller, enough of these being placed underneath the machine to hold it steady while being moved to its proper location. The machines are lifted, and placed upon the skids by means of the bar, lever and bait; the screw jack also playing an important part in this work. The hydraulic jack, or "whisky-jack" as it is called, is also a very useful appliance in millwright

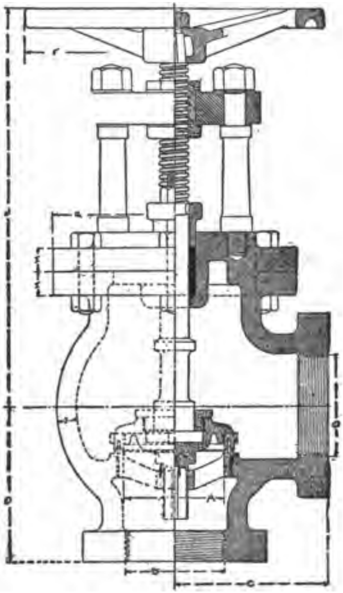


FIG. 129.

VERTICAL SECTION. SCREWED PILLAR YOKE ANGLE VALVE work. A few words of explanation regarding its action may not be amiss here.

The hydraulic jack is a small hydrostatic press made in portable form, the pump being contained inside the plunger, the cylinder forming the body of the jack. Alcohol is used as the liquid for operating the hydrostatic part of the jack. Water may be used instead of alcohol, but it is liable to rust some of the parts, and in cold weather it would freeze, thus causing trouble. To operate one of these jacks, insert the lever so that the shoulder on one side of the lever will project downward and strike a lug on the head of the jack. An ordi-

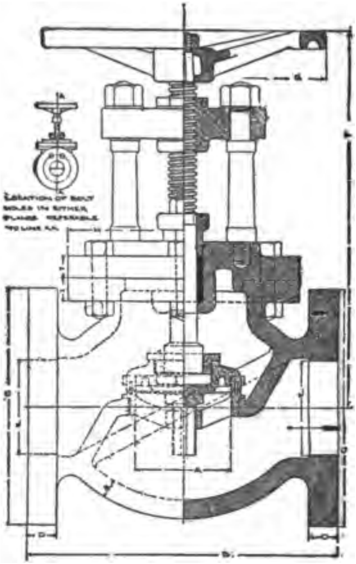

LOCATION OF BOLT HOLES IN EITHER FLANGE PREFERABLE TO LINE A.A.

FIG. 130.

VERTICAL SECTION. FLANGED PILLAR YOKE GLOBE VALVE.

nary pumping movement of the lever will then force the liquid out of the piston into the reservoir underneath it, causing the jack piston and head to rise according to the bulk of liquid displaced. To lower, reverse the lever, turning the lug uppermost, then bear down on the lever and the valve in the pump will be forced to open, allowing the liquid to pass back into the piston reservoir. In some jacks this form of construction is reversed, the piston standing upon the foot or base and the cylinder being attached to the head of the jack. Sometimes the arrangement of the pump lever is opposite that described above, the lug being placed uppermost, and a lifting motion being necessary to lower the jack after the lever has been reversed.

When jacking up machinery every precaution should be observed, and the machine should be followed up closely with substantial blocking in order to guard against any possibility of the accidental dropping of the machine, as much damage might be done, and possibly loss of life occur. When a machine has been safely placed upon skids and rollers, it should not by any means be moved until the pull tackle, and the back haul have both been attached. The pull may be either a differential chain tackle, a heavy rope tackle, or if the machine to be moved is very heavy, a cable and winding drum may be used. For the back haul a heavy rope tackle is preferable, although, in the absence of the tackle, a plain hitch may be made of a heavy rope which is then snubbed around a post or any convenient object that is permanent.

The rigger gang should be under the immediate direction of a competent "boss" or foreman, whose word of command must be the law. The majority of laborers are naturally careless, and are apt to take needless risks

Therefore discretion should be observed in the selection of the rigger gang, and especially in choosing the foreman thereof.

Erecting Machines Upon Foundations. The building of foundations for machines has already been discussed and will not be enlarged upon here, except to say that when making concrete foundations for machinery, use a bank or pit sand whose grains are angular or sharp. If mixed with clay or loam the sand must be washed before using. The gravel used in this work is usually found with the sand. In good gravel the stones are hard, and irregular in shape and size. Broken stone makes better concrete than gravel, but is more expensive. The stone should be graded from ¼ to 1 inch in diameter. The cement used should be of good quality. Mix thoroughly; better have too much water than too little.

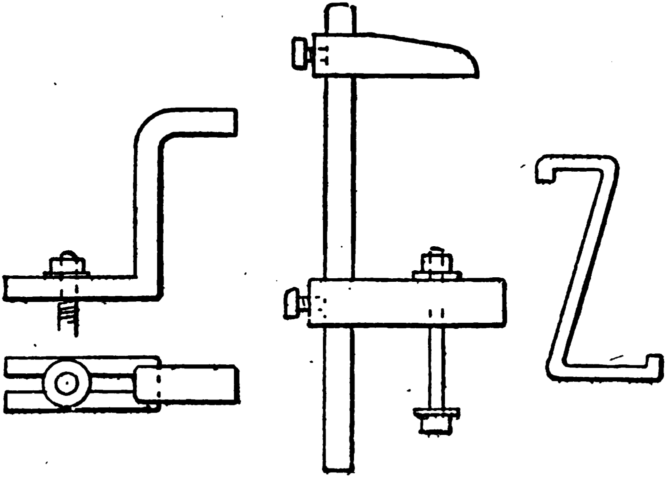


FIG. 131.

If possible, foundations should rest on hard pan, gravel, or hard clay. Made ground should be shoveled out for a depth of at least 12 inches below the bottom of the foundation, and rammed as hard as possible. The 12 inches should then be filled with clay puddle, well rammed in place. Of course the forms must be strong, tight and smooth on the inside. A good form is shown in the cut, Fig. 132. It will be noted that the wall is braced in three places, besides being wired. The wire is about No. 12 iron wire. The braces A are loose, or only tacked in place, and are removed as the wall is built up. Do not nail the sheeting too securely to the studding. Any carpenter can put these forms up and the cheap, unskilled men that wheel the concrete may be employed to assist in the bracing and sheeting. Different methods of bracing the forms will readily suggest themselves to the mechanic in charge, to conform to the local conditions. The cut shows the wall with a footing extending several inches outside it.

Care should be exercised in placing machines upon ready-built foundations to prevent the cracking or crumbling of any portion of the concrete or masonry. The masonry or concrete should be given ample time to set and become solid. Machines generally require to be leveled up after landing them on their foundations, and to do this it becomes necessary to raise one side, or one end, or perhaps just one corner of the base plate a small fraction of an inch, and this usually must be done without the aid of a jack, for the reason that there is no chance for getting the jack underneath the machine, or any part of it that will withstand the pressure.

In such cases a four-sheave tackle should be rigged over head on a beam or girder. A differential chain block will often answer the purpose for light machines.

By the use of a four-sheave tackle, and a winch no strain whatever need be brought to bear upon the foundation by means of jacks and levers.

Tools. A good variety of outdoor, and portable tools are necessary, and should be as handy as possible, so that the kit may not be too clumsy. These should include hammers and cold chisels, a collection of files, round, square and half round; ratchet braces and drills. Portable drill stands for drilling by brace, and which can be used in any position, are shown in Fig. 131. The

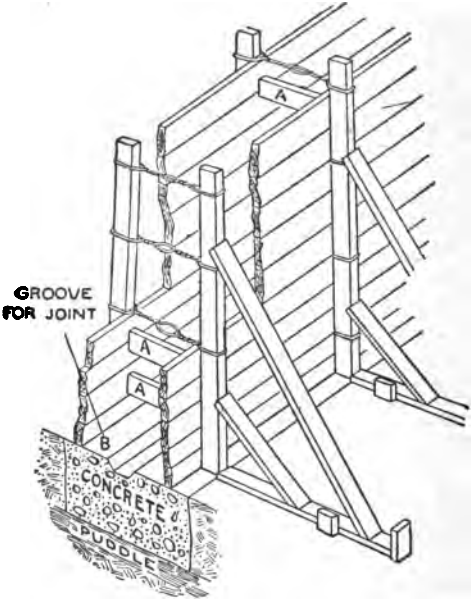


FIG. 132.

FORM OF CONCRETE BASE.

hooked bar shown may be used with a wooden lever having a small center plate.

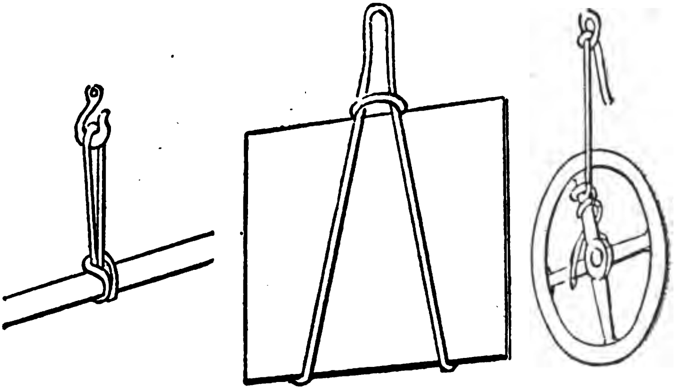


FIG. 133.

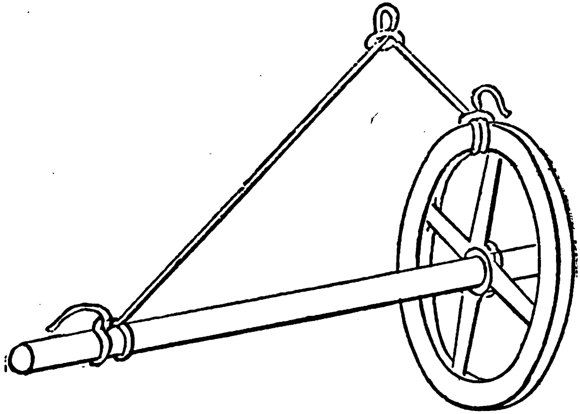


FIG. 133a.

Fig. 133 shows methods of hoisting various parts of machines. Square plates should not be slung by the corners, but across in two places as shown. Fig. 133a shows a method of hoisting a shaft to its position, when pulley is attached. Fig. 133b shows another method of hoisting square plates, also a piston, and a length of shaft.

If brimstone is used for filling under machines, it should be melted in a vessel having a tight fitting cover.

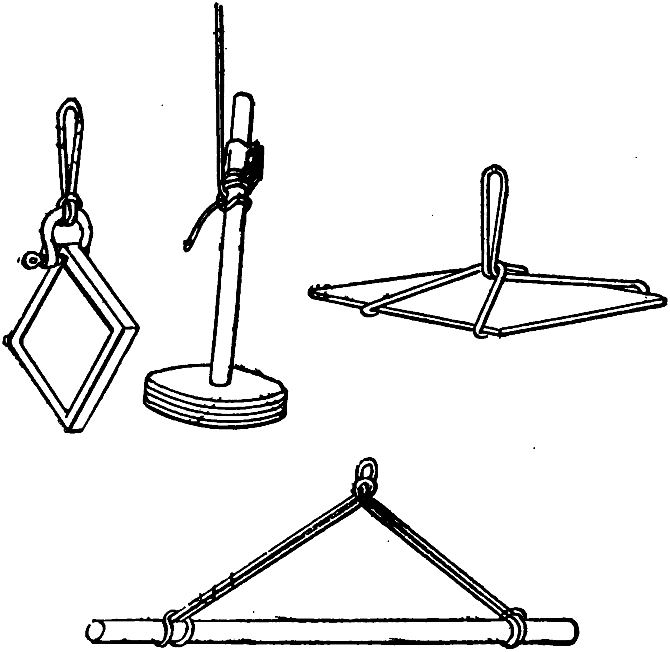


FIG. 133b.

The fire should be light, as brimstone ignites easily, and if allowed to remain over the fire too long after being once melted, it is apt to become waxy, and will not flow readily.

If cement, or plaster of paris is used under a machine, a piece of wire should be used for the purpose of pushing or carrying the running cement into all of the remote corners and cavities in order that they may be filled with cement or plaster, thus securing a firm solid foundation.

Knots and Hitches.—While on the subject of handling, and placing machinery, a short space will be devoted to the discussion and illustration of the various kinds of knots and hitches in use by riggers.

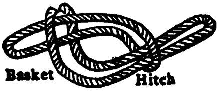


FIG. 134.

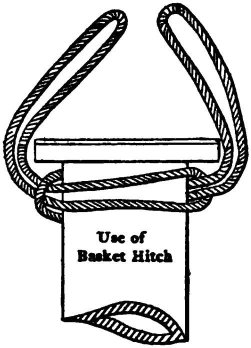


FIG. 135.

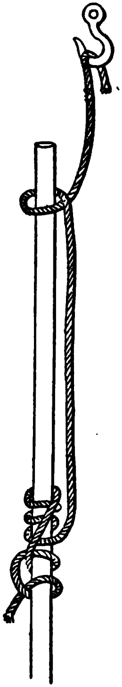


FIG. 136.

Fig. 134 made of two endless slings, and used as shown in Fig. 135, is a reliable and useful basket hitch provided the slings are of equal length, or with one sling just long enough to encircle one-half the circumference of the cylinder, and the other long enough to pass through both loops of the short sling, and have its own loops catch the tackle. A shaft may be hoisted endwise with this hitch by using a collar, or lathe dog for a safety stay, but a better hitch for hoisting a shaft endwise is the biting-rolling hitch shown in Fig. 136. Sometimes when a ready made endless sling is not available, a flat knot is made in a short length of rope and used in place of the sling. The correct form of flat knot is shown at the top of Fig. 137, while the incorrect form, or "granny" knot, is shown below. This latter is unsafe, and should be avoided at all times. Fig. 138 shows a good and safe method of shortening a long rope. This is known as a sheep-shank, and it is evident that any amount, and any length of loop may be used, but it should be remembered that the over-lap at x, x, should be at least 6 inches in length to insure absolute safety.

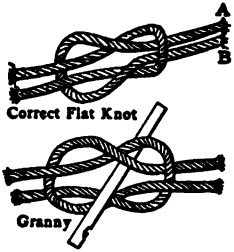


FIG. 137.

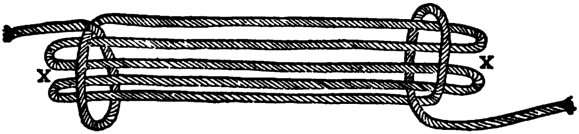


FIG. 138.

The hitch illustrated in Figs. 139 to 142 is made without passing the end of the rope, and provides two loops to which a tackle block may be hooked. Fig. 139 shows the start; Fig. 140, the second stage; Fig. 141, the manner of rolling the two loops into the standing portion of the rope, and Fig. 142 shows the two loops x x brought vertically down (after rolling) and ready for service. The block or fall must be hooked into both loops. This is a safe and reliable hitch that can be wiggled in at any point in a rope, and besides being perfectly reliable, it is easily, and quickly made and unmade.

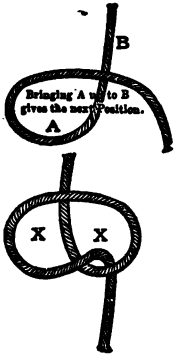


FIG. 139.

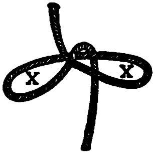


FIG. 140.

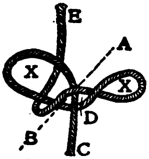


FIG. 141.

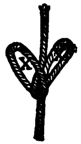


FIG. 142.

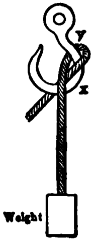


FIG. 143.

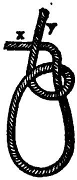


FIG. 144.

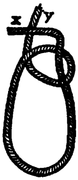


FIG. 145.

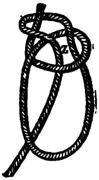


FIG. 146.

Fig. 143 is an old and well-known friend of the rigger. Bow lines are shown in Figs. 144 to 146. In making these knots the slack line x may go either in front of, or back of the standing rope y, as shown in Figs. 144 and 145, but in either case it must be passed through the loop z after going around y. This is shown in Fig. 146, which is an illustration of a completed bow line. Fig. 147 shows how a split hammer handle, or split monkey wrench handle may be temporarily repaired by the endless wound splice. The method of making it up is clearly shown, and it is evident that by pulling at A, the loop B will make a similar loop in D at C, and continued pull will draw the crossed loops out of sight. The loop ends may then be cut off closely. The knot shown completed in Fig. 148 is known as a jury mast knot, also as a "bottle hitch." It can be used in place of a mast iron, at the top of a derrick, or gin pole to make guys fast to.

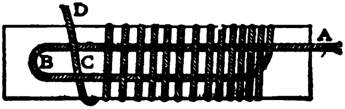


FIG. 147.

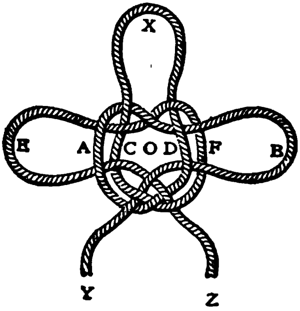


FIG. 148.

It appears at the first glance to be very complicated, but on the contrary it is very easily made. To practice making it, take a piece of strong cord between the thumb and forefinger of each hand, with a space of about 6 inches between the hands. Then twist the cord right-handed with the thumb and forefinger of the right hand only. This will throw up a "bight" like Fig. 149 with the part A under B. Next grasp the loop between the thumb and forefinger of the left hand at the point where the two parts cross. Then move the thumb and forefinger of the right hand along the cord about 6 inches, and throw up another "bight," laying it on top of the first one. This will form Fig. 150. Hold these two "bights" with the left thumb and forefinger, measure off another 6 inches with the right hand, and throw the last "bight." Place it on top of the last one made. This will form a combination, such as shown in

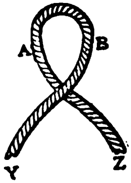


FIG. 149.

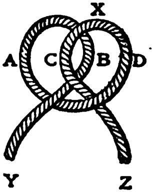


FIG. 150.

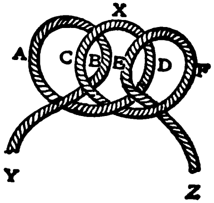


FIG. 151.

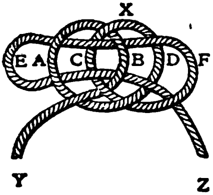


FIG. 152.

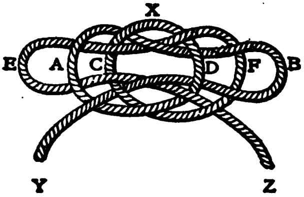


FIG. 153.

Fig. 151. Then take the part E in the last "bight" in Fig. 151 and, while holding the other parts in place, pass it under B, over C, and under A. .

This makes Fig. 152. Then take B, Fig. 152, and pass it under D, and over F, the result being Fig 153. Now while holding E in the left hand, and B in the right, have a helper take hold of x and pull it. This will form the completed knot as shown in Fig. 148. In practice, the central loop O is placed over the reduced top of the mast, or gin pole. The forestay or guy, is made fast to x, the front stays to E, and B, while y, and z form the back stays. Any strain on the stays serves to draw up and tighten O. By pulling on y and z in opposite directions, the knot is easily loosened, and comes out.

Fig. 154 shows how to sling a barrel for hoisting. With both heads of the barrel on, and the bung in place, this is a comparatively easy matter, but with one head out, it is not so easy.

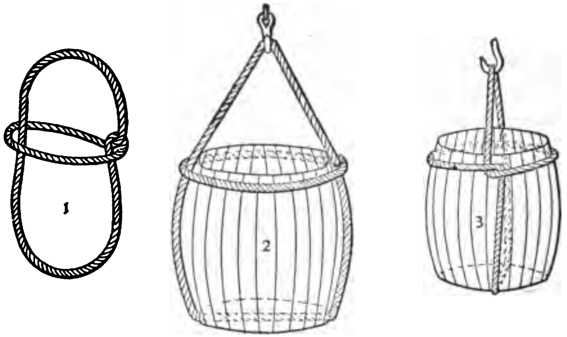


FIG. 154.
SLINGING A BARREL.

The illustrations 1, 2 and 3, in Fig. 154, show how it may be accomplished in a very neat, and easy manner.

Splicing Rope.—The splicing of a rope, especially when used for the transmission of power, is an important matter. The principal points upon which the success of the splice depends are: first, the length of the splice, which depends of course upon the diameter of the rope, and is given in table 36; second, the diameter of the splice, which should be the same as the diameter of the rope; third, the proper securing of the ends of the strands of the splice, which must be fastened in such a manner that they will not wear, or whip out, or cause the overlying strands to unduly wear; fourth, the workmanship of the splice, which should be the best that it is possible to secure.

When splicing an old and a new piece of rope, the new piece should be thoroughly stretched, for, at best, it is an exceedingly difficult task on account of the stretch and difference in diameter of the rope.

The following instructions, and illustrations for making standard rope splices are supplied by the American Manufacturing Company, in their "Blue Book of Rope Transmission." There are many different splices now in use, but the one that experience has proved best is what is known as the English transmission splice.

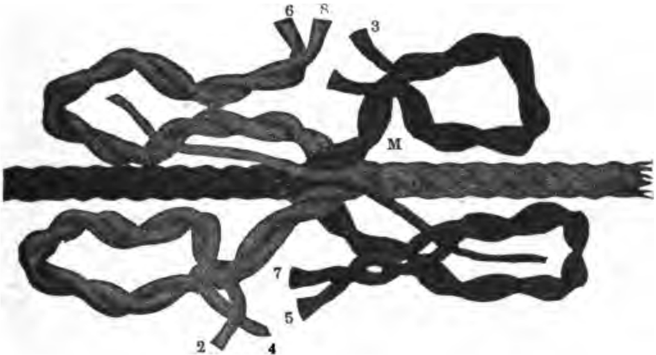


FIG. 155.

In describing this we take for our example a four-strand rope 1¾ inches in diameter, as spliced on sheaves in the multiple system. The rope is first placed around the sheaves, and, with a tackle is stretched and hauled taut. The ends should pass each other from six to seven feet, the passing point being marked with twine on each rope. The rope is then slipped from the sheaves, and allowed to rest on the shafts, to give sufficient slack for making the splice. Unlay the strands in pairs as far back as the twines M, M, Fig. 155, crotch the four pairs of strands thus opened, cores having been drawn out together on the upper side.

Then having removed marking twine M, unlay the two strands 6 and 8 (still in pairs), back a distance of two feet, to A; the strands 1, and 3 (also in pairs), being carefully laid in their place. Next unlay strands 5 and 7 in pairs, to A′, replacing them as before, with strands 2 and 4. The rope is now as shown in Fig. 156. The pair of strands, 6 and 8, are now separated, and 8 is unlaid four feet back to B, a distance of six feet from center, strand 6 being left at A. The pair of strands 1 and 3 having been separated, 3 is left at A, as companion for 6, strand 1 being carefully laid in place of strand 8 until they meet at point B. The two pairs of strands 2-4 and 5-7, are now separated and laid in the same manner, every care being taken, while thus putting the rope together, that the original twist and lay of strand is maintained. The protruding cores are now

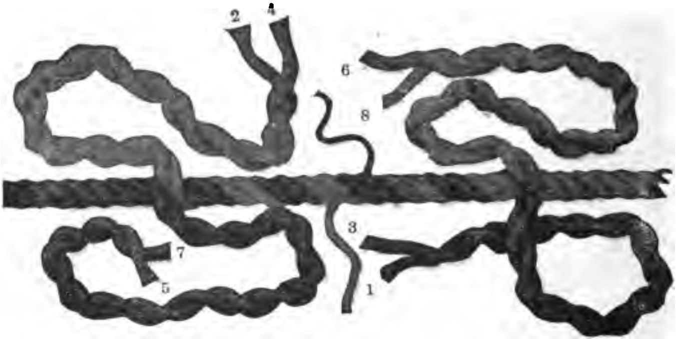


FIG. 156.

cut off so that the ends, when pushed back in rope, butt together. The rope now appears as shown in Fig. 157, and after the eight strands have been cut to convenient

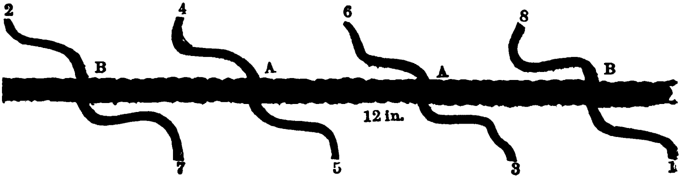


FIG. 157.

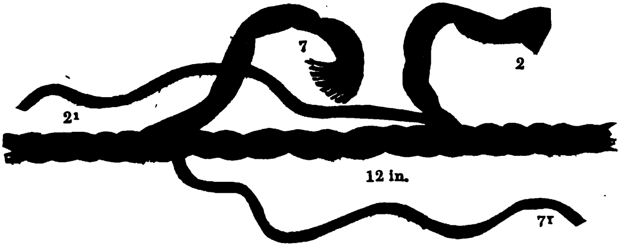


FIG. 158.

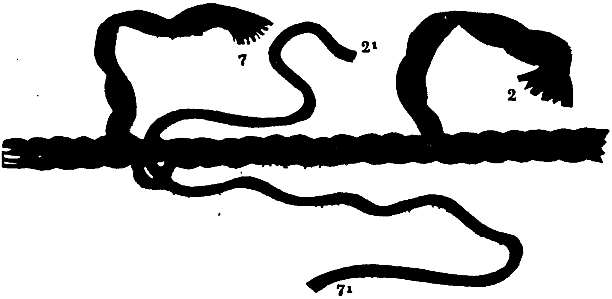


FIG. 159.

working lengths (about two feet), the companion strands are ready to be fastened together and "tucked." This operation is described for strands 2 and 7, the method being identical for the other three pairs.

Unlay 2 and 7 for about twelve to fourteen inches, divide each strand in half by removing its cover yarns, see Fig. 158, whip with twine the ends of interior yarns 2′ and 7′; then leaving cover 2, relay 2′ until near 7 and 7′, here join with simple knot, 2′ and 7′, Fig. 159. Divide cover yarns 7, and pass 2′ through them, continuing on through the rope under the two adjacent strands, avoiding the core, thus locking 2′, Fig. 160. In no event pass 2′ over these or any other strands.

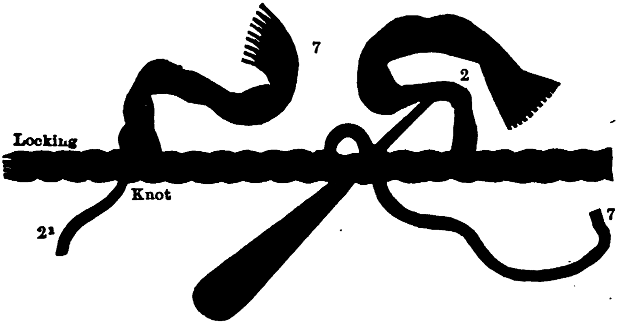


FIG. 160.

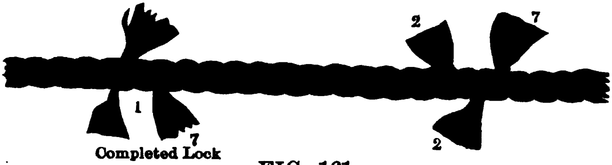


FIG. 161.

Half strand 7′ must now be taken care of. At the right of the knot made with 2′ and 7′, 2′ is slightly raised with a marlin spike, and 7′ passed or tucked around it two or three times, these two half strands forming in this way a whole strand.

Half strand 7′ is tucked until cover 2 is reached, whose yarns are divided, and 7′ is passed through them, and drawn under the two adjacent strands, forming again the lock.

The strand ends at both locks are now cut off, leaving about two inches, so that the yarns may draw slightly without unlocking. This completes the joining of one pair of strands. See Fig. 161. The three remaining pairs of strands are joined in the same manner.

After the rope has been in service a few days, the projecting ends at locks wear away, and if tucks have been carefully made, and the original twist of yarns preserved, the diameter of the rope will not be increased, nor can the splice be located when the rope is in motion.

As before stated, Table 36 which follows, contains useful data regarding the splicing of manilla transmission rope, also approximate weight in pounds per lineal foot, of rope of various diameters, together with the breaking strength, maximum allowable tension in pounds, and smallest diameter of sheaves over which the different sizes of rope may be run with safety.

TABLE 36.—MANILLA TRANSMISSION ROPE. SPLICING AND SHEAVES.

Diameter of Rope in Inches	Square of Diameter	Approximate Weight per Foot, Pounds	Breaking Strength, Pounds	Maximum Allowable Tension, Pounds	Length of Splice in Feet			Smallest Diameter of Sheaves in Inches	Maximum Number of Revolutions per Minute
					3-Strand	4-Strand	6-Strand		
½	.25	.12	1750	50	6			20	1060
9/16	.2906	.16	2730	80	6			24	970
¾	.5625	.20	3950	112	6	8		27	760
⅞	.7656	.26	5400	153	6	8		32	650
1	1.	.34	7000	200	7	10	14	36	570
1⅛	1.2656	.43	8900	253	7	10	16	40	510
1¼	1.5625	.63	10,900	312	7	10	16	45	460
1½	2.25	.77	15,700	450	8	12	18	54	380
1¾	3.0625	1.04	21,400	612	8	12	18	63	330
2	4.	1.36	28,000	800	9	14	20	72	290
2¼	5.0625	1.73	35,400	1012	9	14	20	81	255
2½	6.25	2.13	43,700	1250	10	16	22	90	230

SHIFTING BELTS AND BELT SHIFTERS.

In all shops and mills the nature of the work requires more or less stopping, and starting of the various machines, and in many cases the reversal of the same. This necessitates shifting belts, and loose-pulleys, together with belt shifters, the reversal of the machines being accomplished by crossed belts.

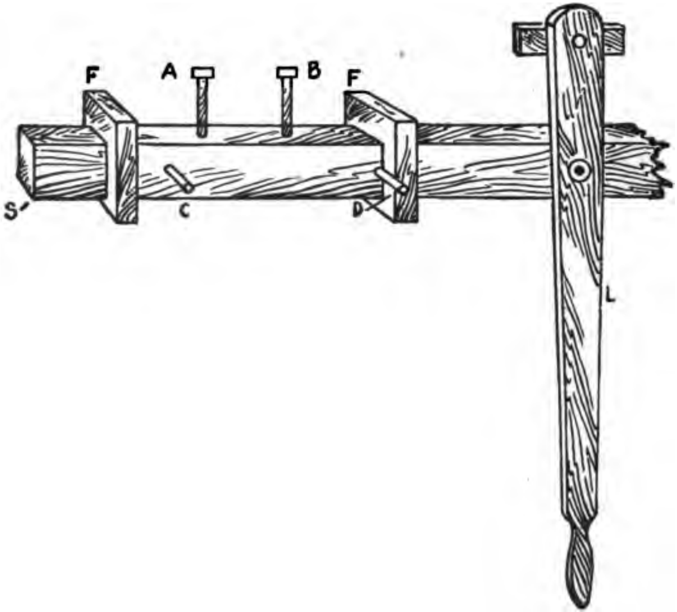


FIG. 162.

Shifting belts are not so much in vogue now as they formerly were, they having been to a great extent displaced by the rim-friction clutch, a very useful and convenient device. Nevertheless shifting belts are still much in use, and as long as they continue to be used, just so long will belt shifters be required, therefore a few words on this subject will no doubt be appreciated. In the first place a belt shifter should not touch the belt except during the actual time occupied in shifting the belt from one pulley to the other. This means that whether the belt is running on the tight pulley, or on the loose pulley, the shifter should not touch the belt, nor be made, as too many of them are, to do duty as a belt guide to keep the belt on the pulley. When a belt requires a guide to keep it running fair on the pulleys, it is a pretty fair indication that the shafting is out of alignment. Fig. 162 shows the construction of an ordinary belt shifter. The piece of 2 x 4 inch scantling S is supported in the guides E and F, through which it is free to move back and forth.

The pins C and D, prevent too much movement of the shifter bar S, while the lag screws A and B, serve to force the belt from one pulley to the other when the bar is moved endwise.

The lever L for moving the shifter bar is made of sufficient length to reach to the floor, or any other point most convenient for operating it. The lag screws, A and B should be a sufficient distance apart to allow a clearance of at least one half inch between them and the edges of the belt for all belts of four inches or less in width, and for all belts wider than four inches there should be an increased amount of clearance between belt and guide pins.

If instead of lag screws, three quarter inch bolts are used for guide pins, much better results will be obtained. These bolts should pass entirely through the shifter bar S, and have a nut above the bar, and another nut underneath. They may then be securely tightened in their position with no danger of becoming loosened. At C and D are shown pins which are for the purpose of preventing the shifter bar from being moved too far in either direction. When the belt has been shifted to the desired position, either on or off the tight pulley, lever L is secured in that position and thus retains the shifter in its proper location. Fig. 162 shows but one of the numerous devices used for changing belts from tight to loose pulleys, and vice versa. The appliances hitherto referred to are of course adapted exclusively to belts running off horizontal shafts. Belts cannot be shifted in this manner on pulleys running on vertical shafts, as for instance with the vertical spindles in flouring mills, owing to the fact that the weight of the belt tends to make it slide down the face of the pulley, and this tendency must be opposed by flanged pulleys. Moreover, when several belts are led off from the main vertical shaft to

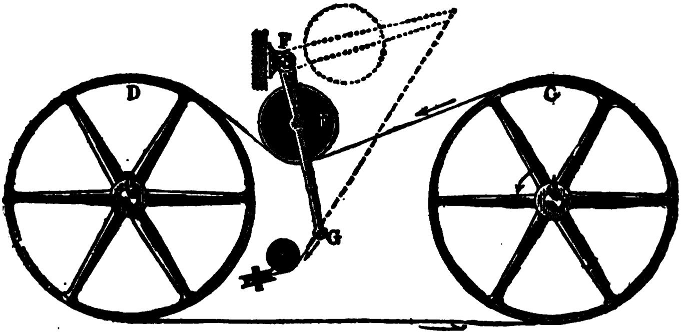


FIG. 163.

separate spindles, there is generally not sufficient space to arrange two pulleys for each belt. It is the usual custom in such cases to effect the engagement, and disengagement of the belt by means of tightening pulleys. When these act on the ordinarily slack belt they produce sufficient friction between the pulley faces and the belt to transmit the motion. Fig. 163 shows such an arrange-

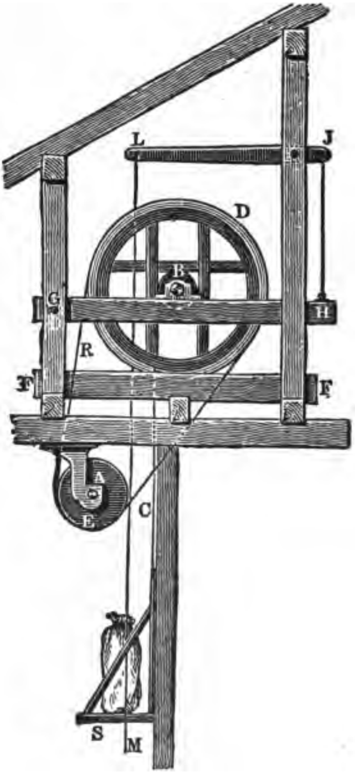


FIG. 164.

ment. An examination of the cut will show that the tightening pulley is pressed against the slack, or driven side of the belt. With this arrangement the force with which the tightening pulley is pressed against the belt is less, and therefore the journal friction of the pulley is less than when the tightener is pressed against the tight side of the belt. In some cases the belt is tightened by temporarily increasing the distance between the shafts on which are the two pulleys connected by the belt, the driven shaft being the one usually shifted.

One of the best known examples of this kind is to be found in the hoists of flouring mills.

A hoist of this kind is shown in Fig. 164, in which B is a drum located in the upper story of the building, while C is a belt or rope from which the platform S is suspended, and which winds on to the drum when the latter has a right handed rotation. This drum B can be driven by a belt connecting the pulley E on the driving shaft A with a larger pulley D, on the hoisting shaft carrying the drum B, provided the belt receives the necessary tension. In order that this tension may be imparted to the belt at any time, the bearing of the hoisting shaft is fastened to a beam G, H, pivoted at G, and by means of the two armed lever L J the bearing of B can be lifted by simply pulling on the cord M which extends to the lower floor of the mill. Thus the usually slack belt R is tightened, and the hoisting drum B is turned as long as there is a pull on the rope M. When the pull ceases, all motion ends, and the platform S does not sink, owing to the fact that the friction of the two lateral brakes F, F, prevents a reversal of the motion of pulley D. Not until a moderate pull on the rope M has lifted the hoisting drum sufficiently to release the friction between pulley D and the brakes, without however, tightening the driving belt R, does the platform S either with or without its load sink by its own weight with a velocity regulated by the brake F F.

Removing Pulleys from Shafting. When a solid pulley is to be removed from a section of shaft, sledge hammers should not be used on the hub or arms, as cast iron pulleys are very liable to break or crack under repeated blows.

One method of removing a pulley is illustrated in Fig. 165, and may be applied when a location can be found between two solid walls that will admit the necessary apparatus. As will be observed, the method consists in forcing the shaft through the pulley by means of a screw jack pressing against the end of the shaft, the pulley itself being held stationary by a piece of pipe that is slipped over the shaft, and rests against the opposite wall. Another method of removing a pulley is illustrated in Fig. 166, in which the piece of shaft holding the pulley is mounted on an incline, with the lower end resting against a solid wall, while another piece of old

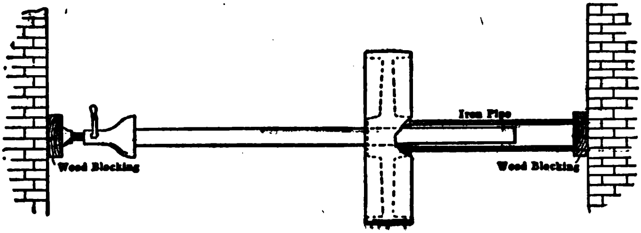


FIG. 165.

shafting is used against the hub of the pulley, as a ram. In order to prevent damaging the pulley hub,

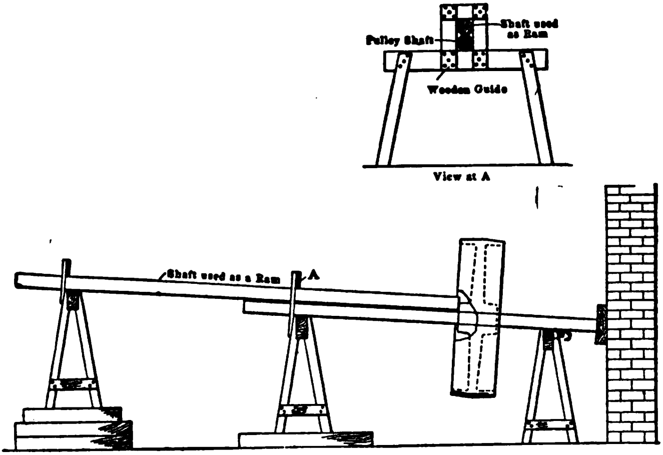


FIG. 166.

and also to have its force applied to the best advantage, the ram should be used in a direct line with the direction

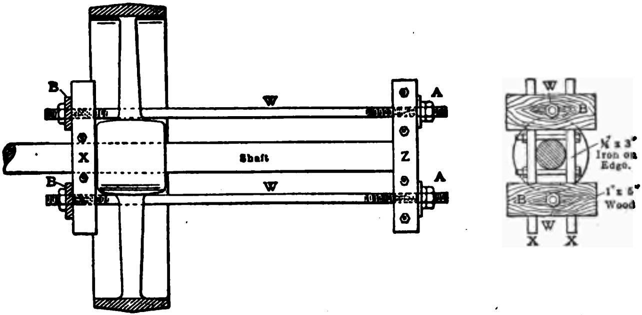


FIG. 167.

of removal. Fig. 166, is self-explanatory. A very convenient method that may be applied to any size pulley provided the bolts W W are large enough, is shown in

Fig. 167. The bolts W W must have long threads cut on them. Pressure is brought to bear on the hub by pulling up on the nuts A, A, each of which should be drawn a little at a time in order to keep the strain equalized between the two bolts. In case the pulley comes very hard, it may be assisted when the strain is on the bolts, by blows from a sledge hammer delivered on the block x. This method can be used only in cases where the end of the shaft can be reached as shown in the illustration.

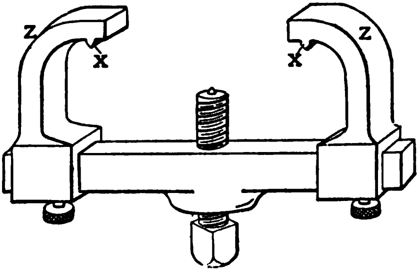


FIG. 168.

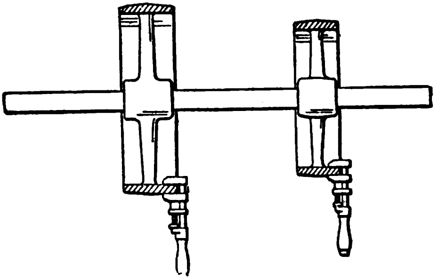


FIG. 169.

For removing pulleys from electric generators, and motors, the device shown in Fig. 168 is very convenient provided the pulleys are near the end of the shaft, and they generally are with those machines. The arms z, z, may be adjusted so as to take hold of the hub, or the arms of the pulley, while the screw applied to the center of the shaft brings the pressure to bear.

After a pulley is once loosened, a good way to run it off the shaft is to apply a monkey wrench to the rim, and keep turning the pulley around, while the shaft is held from turning by using another monkey wrench on the rim of one of the tight pulleys. This method is illustrated in Fig. 169.

Keys and Set Screws. Concerning the dimensions of keys, Prof. Coleman Sellers says: "The size of keys, both for shafting, and for machine, tools are the proportions adopted by William Sellers and Company, and rigidly adhered to during a period of nearly forty years. Their practice in making keys, and fitting them is, that the keys shall always bind tight sidewise, but not on top and bottom; that is not necessarily touch either at the bottom of the key seat in the shaft, or touch the top of the slot cut in the gear wheel that is fastened to the shaft; but in practice, keys used in this manner depend upon the fit of the wheel upon the shaft being a forcing fit, or a fit that is so tight as to require screw pressure to put the wheel in place upon the shaft." Tables 36a and 36b give the sizes of keys for shafting, and for machine tools respectively.

Diameter of Shaft, in.	Size of Key, in.
$1\frac{1}{4}$ $1\frac{7}{16}$ $1\frac{11}{16}$	$\frac{5}{16} \times \frac{3}{8}$
$1\frac{15}{16}$ $2\frac{3}{16}$	$\frac{7}{16} \times \frac{1}{2}$
$2\frac{7}{16}$	$\frac{9}{16} \times \frac{5}{8}$
$2\frac{11}{16}$ $2\frac{15}{16}$ $3\frac{3}{16}$ $3\frac{7}{16}$	$\frac{11}{16} \times \frac{3}{4}$
$3\frac{15}{16}$ $4\frac{7}{16}$ $4\frac{15}{16}$	$\frac{13}{16} \times \frac{7}{8}$
$5\frac{7}{16}$ $5\frac{15}{16}$ $6\frac{7}{16}$	$\frac{15}{16} \times 1$
$6\frac{15}{16}$ $7\frac{7}{16}$ $7\frac{15}{16}$ $8\frac{7}{16}$ $8\frac{15}{16}$	$1\frac{1}{16} \times 1\frac{1}{8}$

Length of key-seat for coupling = $1\frac{1}{2} \times$ nominal diameter of shaft.

TABLE 36^a
Size of Keys for Shafting

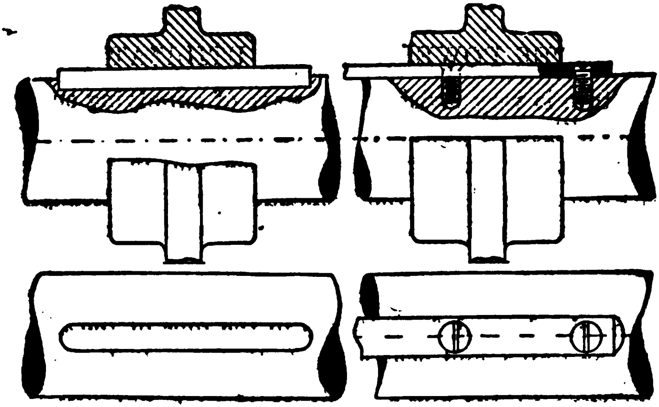


FIG. 170.

FIXED FEATHER-KEYS.

Diam. of Shaft, in.	Size of Key, in. sq.	Diam. of Shaft, in.	Size of Key, in. sq.
$\frac{15}{16}$ and under	$\frac{1}{8}$	4 to $5\frac{7}{16}$	$\frac{13}{16}$
1 to $1\frac{3}{16}$	$\frac{3}{16}$	$5\frac{1}{2}$ to $6\frac{15}{16}$	$\frac{15}{16}$
$1\frac{1}{4}$ to $1\frac{7}{16}$	$\frac{1}{4}$	7 to $8\frac{15}{16}$	$1\frac{1}{16}$
$1\frac{1}{2}$ to $1\frac{11}{16}$	$\frac{5}{16}$	9 to $10\frac{15}{16}$	$1\frac{3}{16}$
$1\frac{3}{4}$ to $2\frac{3}{16}$	$\frac{7}{16}$	11 to $12\frac{15}{16}$	$1\frac{5}{16}$
$2\frac{1}{4}$ to $2\frac{11}{16}$	$\frac{9}{16}$	13 to $14\frac{15}{16}$	$1\frac{7}{16}$
$2\frac{3}{4}$ to $3\frac{15}{16}$	$\frac{11}{16}$		

TABLE 36^b
Size of Keys for Machine Tools

Regarding the holding power of set screws, as tested by G. Lanza, the results obtained were as follows: The set-screws used were of wrought-iron, $\frac{5}{8}$ of an inch in diameter, and ten threads to the inch; the shaft used was of steel and rather hard, the set-screws making but little impression upon it. They were set up with a force of 75 lbs. at the end of a ten-inch monkey-wrench. The set-screws used were of four kinds, marked respectively A, B, C, and D. The results were as follows:

A, ends perfectly flat, 5/16-in. diameter......1412 to 2294 lbs.; average 2064
B, radius of rounded ends about ½ inch..2747 to 3079 lbs.; average 2912
C, radius of rounded ends about ¼ inch..1902 to 3079 lbs.; average 2573
D ends cup-shaped and case-hardened....1962 to 2958 lbs ; average 2470

WATER POWER AND ITS UTILIZATION.

Nature is very lavish in her ways of supplying energy to be utilized as power in various forms for the welfare and advancement of the human race, and in none does she excel more than in the tremendous and practically inexhaustible supply of energy that is stored in water seeking its level.

That never-ending round of circulation from the sea to the clouds, and from thence to the tops of the mountains and hills, and back again to the sea, which water is compelled by the sun's influence to keep up, will continue to be a source of power when coal shall be forgotten. As just one instance of the vast amount of power available from this source, it is said that if all the daily output of coal in the world could be used in making steam to drive pumps, it would hardly suffice to pump back the water flowing down the Niagara River.

The efforts being put forth by the United States Government for the conservation of its forests and watercourses cannot be too highly commended, and should receive the hearty indorsement of all.

Water Wheel Setting.—The development of the modern turbine water wheel, with its draft tube has wrought a wonderful change in methods of wheel setting so far as the millwright is concerned. Whereas, it was once the height of the millwright's ambition to be able to frame and erect a deck flume in which water could be confined under 10 to 40 feet head, and not a drop of water found leaking from the flume, nowadays the wooden flume, the framed penstock and the wooden wheel case are things of the past and are seldom seen, much less constructed. The boiler maker and the concrete worker do about all the water-wheel setting now, and it is done to stay. The old-time wooden construction used to last about seven years before it came time to repair it, and at ten years the whole thing had to be replaced with new material.

The open flume, the decked flume and the penstock have all been replaced by steel and cement so thoroughly that wooden flume making is an obsolete branch of millwrighting.

Modern methods have reduced the setting of a turbine wheel from a task of several weeks for a considerable force of skilled men, to the work of a few days for a foreman, a gang of laborers, and a couple of boilermakers, the latter to rivet the wheel case to the steel penstock, and draft tube. The foundation for the wheel case was made when the foundations of the mill were constructed, and all that can be seen of it are several anchor bolts projecting from a smooth concrete floor. So exact has been the engineering work with transit and station-rod, that after the wheel case has been dropped in place, it needs nothing except being twisted around a little to coincide with the shaft and building lines and perhaps leveled up a bit with some brimstone and a steel wedge or two.

The millwright in all probability will be called upon to exercise a general supervision over the work of setting the wheel case, and also of the construction of the concrete flume, or steel pipe line, as the case may be, but he has the blue prints, and specifications of the manufacturers to guide him in this work, and is expected to follow these.

Shaft Supports.—In order that the highest efficiency may be obtained from water wheels it is absolutely necessary that their shafts be properly supported, as regards both boxing and timbering. These are two very important factors in the proper maintenance of a water wheel.

Penstocks.—Steel, or concrete are the best materials for penstock building, unless the mill is situated in a locality where suitable timber is abundant.

Concrete should not be used for building penstocks situated above ground, for the reason that a penstock is always under a tensile strain, and concrete members should not be subjected to such strains. In fact the millwright may be at different times called upon to construct penstocks of various kinds of material, even brick being used in some cases, the kind of material used depending upon the locality, and the surroundings.

THE TURBINE.

In view of the rapidly increasing use of the water turbine as a prime mover, it may not be out of place to devote a short space to an explanation of the principles controlling its design, and the laws governing its action. The turbine is a form of water wheel which makes use of the energy developed by water as it flows between curved vanes or channels, which, owing to their shape, so deflect, and alter the course of the water in its passage that it exerts a reactionary force, thereby causing and maintaining motion. This action is an application of Newton's second, and third laws of motion, as will be readily seen.

Referring to Fig. 171, let A B represent a vane moving in the direction indicated by the arrow, with a uniform velocity c; let O A represent the direction and magnitude, v of a thin stream of water entering at A in such a manner that there is no sudden change of velocity, or direction of flow. In order that this may be accomplished, it is required that the parallelogram, Ov Ac, must be the parallelogram of velocity of the water, A C being the tangent to the vane at A and Ov, Oc must represent v, the velocity of the water relative to the vane, and c the velocity of the vane.

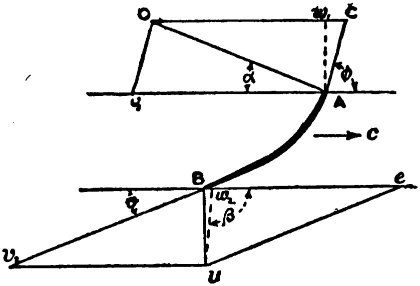


FIG. 171.

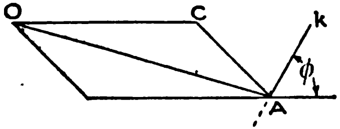


FIG. 172.

If, after drawing A O, O C (Fig. 172), C A were not a tangent to the vane as Ak at A, then the water would not glide along the vane without shock, but its direction and velocity would be suddenly altered at A. We must leave this latter case at present, and return to Fig. 171. A perpendicular Aw_1 upon O C gives us the tangential velocity, or velocity of whirl $Ow_1 = w_1$ of the water at entry; the angles Φ and α may also be noticed.

Again, at the point B, where the water leaves the vane, $Bv_2 = v_2$ is the relative velocity, $Be = c$, $Bu = u$ the total velocity, and Bw_2 is the velocity of whirl at discharge, Bw_2u being a right angle, and the angles Φ and β should be noted. Let W be the weight in pounds of water passing in time t, and let P be the component parallel to the direction of motion of the vane, of the reaction between water and vane, then $\frac{W}{g}(w_1 - w_2) = Pt$, because the momentum is changed from $\frac{Ww_1}{g}$ to $\frac{Ww_2}{g}$, in time t by a force P. Hence the work done in time t on the vane $= ct \times P$ foot pounds; $= c \times Pt = \frac{Wc}{g}(w_1 - w_2)$, a quantity independent of t. The work done per pound $= \frac{c}{g}(w_1 - w_2)$ ....(7)

If entry is to take place without shock, and certain velocities and angles are assumed, it is clear that by construction, or trigonometry the remaining velocities and angles can be found.

For example, if β θ and v_2 are assumed, then u and c are found by the parallelogram βv_2 u e. Again, if v = O A, is known with a and c, the parallelogram Ac Ov will make it possible to find v, and θ.

However useful graphical methods may be in saving calculation, it will be found that very little more time is required and greater accuracy is obtained by the use of a slide rule and a table of sines, cosines, tangents, etc. Also, it is quite possible that calculations must be made when a drawing board is not easily obtainable, and in such cases the following equations will prove useful:

$$\left.\begin{array}{r}W_1=V\cos\alpha\\ C-W_1=V_1\cos\theta\\ C\sin\theta=V\sin(\alpha+\theta)\\ V_1\sin\theta=V\sin\alpha\end{array}\right\}\ .(8)\qquad \left.\begin{array}{r}W_2=U\cos\beta\\ C-W_2=V_2\cos\theta\\ C\sin\theta=U\sin(\beta+\theta)\\ V_2\sin\theta=U\sin\beta\end{array}\right\}\ldots.(9)$$

Assuming, as before, B, θ, and v_2 known, then the fourth equation of (9) gives u, and the first will give W_2, and c may be found from the second or third. Again, if v, a and c are known w_1 is found from the first of equations (8); Φ may be calculated from the third, which may be written $c - w_1 = v \cos\theta \sin a$, and v_1 may now be obtained from the second or fourth.

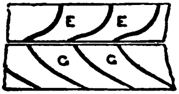


FIG. 173.

The equations (8) are unaltered if Φ be greater than a right angle; and the above reasoning will apply equally well when the vane moves, not in a straight line, but in a circle whose plane is perpendicular to the paper, the axis of rotation being then parallel to the paper. The student will now be able to understand the axial, or parallel flow turbine, through which the water flows in a direction parallel to the axis of rotation.

Figs. 173 and 174 are sectional elevations; the former is part of a cylindrical developed section taken through the guide vanes E and the wheel vanes G, the direction of motion of G being to the left, while the guide vanes are fixed. The water, collected in a reservoir above, flows downwards into the guide apparatus, in which it

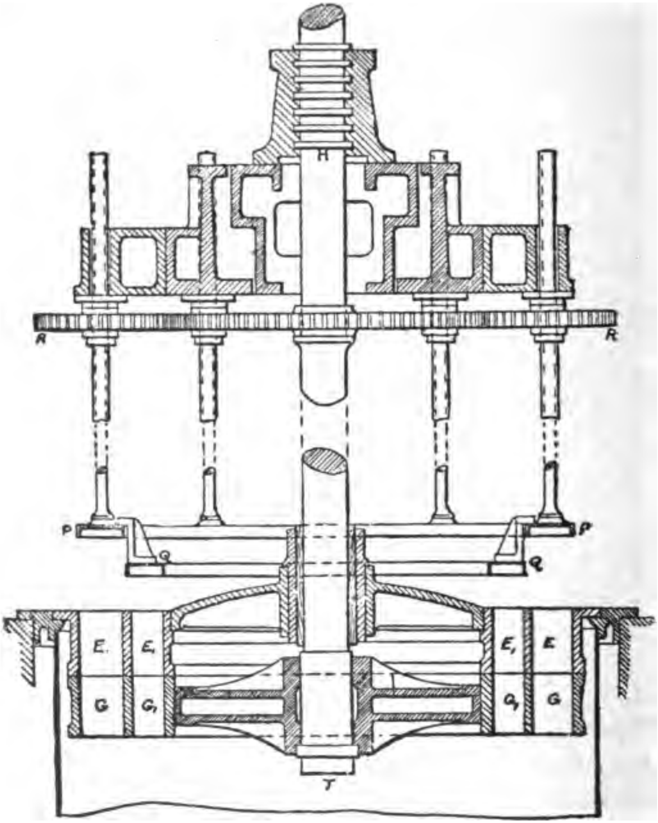


FIG. 174.

is given a forward velocity v; leaving these, it should enter the wheel G without shock, and flow out vertically, after it has imparted some of its energy to the wheel in consequence of the change of velocity of whirl from w_1 to w_2 which should be zero, the reason for which will be explained later on. Fig. 174 is a vertical section through the axis of rotation.

The turbine of which Fig. 174 is a vertical section, is 9 feet 6 inches in diameter (outside), and is carried on a vertical wrought iron shaft 8 inches in diameter. The head of water varies from 12 feet to 16 feet, according to the condition of the river. It is capable of developing about 250 H. P. when using 200 cubic feet of water per second, with a head of 12 feet, showing an efficiency of 91.6 per cent, which is certainly remarkable, when we bear in mind that 85 per cent is considered very good. During periods of flood the quantity of the flow can be increased to about 280 cubic feet per second. The average number of revolutions is 48 per minute, giving a circumferential speed of 24 feet per second. This turbine wheel is in two parts, and is termed a double axial turbine. When the water in the river is low, there is a greater difference of level between the upper and lower water, thereby increasing the head, and during this period the outer rings, E, G, of the guide vanes and wheel are used, the inner rings, E_1, G_1, being closed. But when the water is high in the river and the fall not so great, the lower level being raised, both rings of the turbine are opened. In this manner a large volume of water with a short fall can be used, and vice versa a smaller volume with a higher fall. The center of the upper directing portion, E, of each turbine is closed by a fixed bill, in which is provided a bearing lined with strips of wood, for the turbine shaft. The shaft is carried by a compound collar bearing H, at the upper end, from which the entire weight of the turbine is suspended. This bearing is carried by a pair of cast iron girders, which are fixed across the turbine house, resting upon the side walls.

Another bearing between H and the rings, P Q, is also given, supported by a girder. This bearing is also lined with wood strips, and above it a collar is attached to the turbine axle with clamping screws, and is adjusted so as to be just clear of the bearing in ordinary working; but when the water pressure becomes very great a slight settlement of the turbine shaft takes place, and this collar then takes a bearing, and supports part of the weight so as to prevent there being too great friction at H. The two rings, P Q, as before mentioned, regulate the supply of water. The outer ring P is suspended by six spindles, which are screwed at the top through nuts in the spur wheels R. These are geared together by a large center wheel so that they can be all turned together by gearing from a hand-wheel (not shown in the figure), so that P can be raised or lowered to adjust the opening for the passage of the water to the outer ring. Q is raised by screwing up the outer ring until it catches projecting brackets fixed to Q, which then rises with P. As usual, the power is taken from the shaft by means of a bevel wheel above H. It will be noticed that a suction tube T is used, reaching about 4½ ft. below the bottom of the wheel, so as to allow for variations in the lower level of the stream. If this were not done, head would be lost when the water is low in the river.

The theory of the action of the suction tube will be explained later on; suffice it to say here, that so long as the lower level of the water is above the bottom of the suction tube, the turbine may be placed at some height above the tail race, without any alteration of the effective head. Thus the lengths of the shaft and the six spindles may be less than they would otherwise be, the reduction of weight of the former tending to decrease the loss by friction at H.

As has already been stated, the velocity of whirl at discharge should be zero—that is, that the water should flow out axially; or, in this case, vertically. Now u (Fig. 171) is the total velocity of outflow, but u sin β is the component of the velocity that carries the water out of the wheel. However great w_2, the other component, may be, no more water will flow through a wheel of a given size so long as u sin β is fixed. Thus, for a given wheel and given quantity of water, u sin β must have a fixed value; but u must be as small as possible, because one of the losses of head is $\frac{u^2}{2g}$, the energy in each pound of water as it leaves the wheel; but u is never less than u sin β, so that the least value of u will be when $\beta = 90$ deg., and therefore $w_2 = 0$.

From equation (7) the work done per pound

$$= \frac{c}{g} (w_1 - w_2)$$

$$= \frac{c}{g} w_1 \text{ when } w_2 = u \cos \beta.$$

Neglecting friction, the formula for ascertaining the H. P. follows: Let Q = number of cubic feet of water per second,

$$\text{then H. P.} = \frac{Qcw_1 \times 62.5}{g \times 550} \quad \ldots\ldots\ldots\ldots\ldots (10)$$

and the maximum hydraulic efficiency is

$$\eta = \frac{cw_1}{gH} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (11)$$

a quantity which varies between .65 and .9 when the turbine is running at the right speed, and none of the guide passages is closed, either wholly or partially, as this reduces the efficiency.

The ordinary parallel flow turbine is not divided into two parts as in Fig. 174, and c is then the velocity at the mean radius of the wheel vanes. Referring to Fig 174, let c, C be the velocities at the mean radii of the blades G, G_1, and w_1, W_1, the corresponding velocities of whirl q_1, Q_1, the quantities of water flowing per second through E, E_1, so that $q_1 + Q_1 = Q$ cu. ft.

$$\text{then H. P.} = \frac{62.5 \ (q_1 cw_1 + Q_1 CW_1}{550g}$$

and hydraulic efficiency

$$\eta = \frac{q_1 \ cw_1 + Q_1 CW_1}{g\,QH}$$

Classification of Turbines.—All turbines belong to one of two classes, called reaction and impulse turbines. In the former, when working at full power, all the guide and wheel passages are filled with water, and the turbine is said to be drowned, and the velocity of flow in one part can be determined when that in any other part is known; or, to put it mathematically, if A_1, A_2, A_3 be the cross-sections of the stream at any point, and v_1, v_2, v_3 the velocities perpendicular to those sections, then $v_1 \ A_1 = v_2 \ A_2 = v_3 \ A_3$.

Wherever the quantity of water is large, and the fall moderate, a reaction turbine is generally used.

An impulse turbine is not drowned, the buckets are not filled, and in some cases, in which there is only partial admission, each bucket is empty during part of a revolution. Air is required in the buckets, so that the pressure may always be that of the atmosphere, ventilating apertures being made in their sides for this purpose. A suction tube cannot be used with this class of turbine, which is suitable for high falls and moderate or small quantities of water.

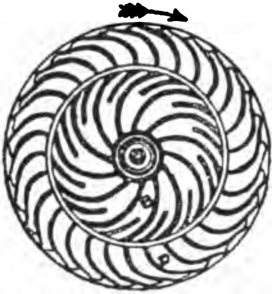


FIG. 175.

Thus a reaction turbine may be made for a fall of 14 feet, and 200 cubic feet of water per second; but an impulse turbine is required for a fall of 600 feet, and 20 cubic feet of water per second. In the former case an impulse turbine might be used in place of the reaction turbine, but the high number of revolutions required for the reaction turbine in the latter case, where the fall is high, makes it necessary to use an impulse turbine; for with a small quantity of water, a turbine that is filled must be of small diameter, and as the velocity of rotation at the radius at which inflow takes place is not less than .45 $\sqrt{2gH}$ for a reaction turbine, and is generally greater, the number of revolutions would become inconveniently high when H is large.

Reaction and impulse turbines may be again divided into three classes: radial, axial, combined or mixed flow. In the first the water flows outwards or inwards; the second has been already described; and in the third the water enters approximately radially, and leaves axially. The axes may be in all cases vertical or horizontal. Fig. 175 shows a section perpendicular to the axis, and through the guide apparatus and wheel of a radial outward flow or Fourneyron turbine. At B are the guide vanes and at D the wheel vanes, the arrow showing the

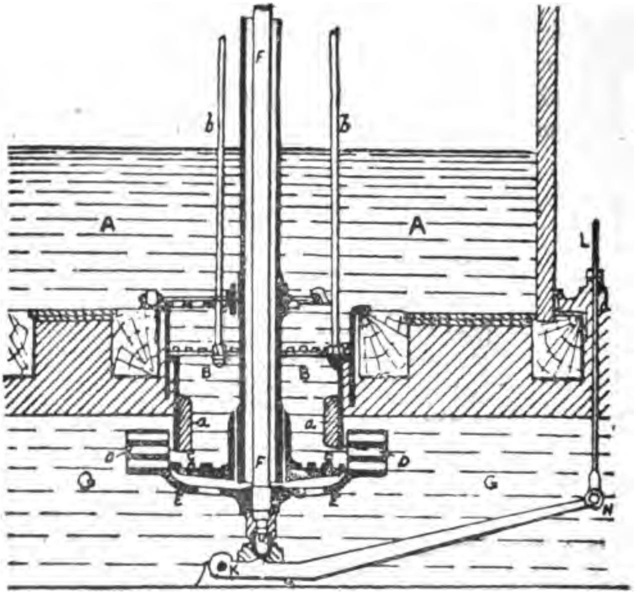


FIG. 176.

direction of rotation. Fig. 176 is a vertical section of this type of wheel. A is the penstock, B the supply cylinder, which consists of two concentric tubes; the upper one being fixed, while the lower slides within it like the inner tube of a telescope, and is raised or lowered by the rods b. Near the upper edge of the inner tube is a leather collar making a watertight joint between it and the outer tube. The lower part, a, of the inner tube acts as a regulating sluice for all the orifices at once. It has fixed to its internal surface wooden blocks, shaped so as to round off the turns in the course of the water towards the orifices.

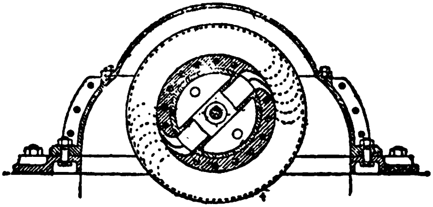


FIG. 177.

The bottom of the supply cylinder is formed by a fixed disc, C, which is supported by hanging at the lower end of a fixed vertical tube enclosing the shaft. This disc carries the guide blades. D D are the vanes of the wheel, which in this case are divided into three sets, or horizontal layers, by two intermediate crowns, or horizontal ring-shaped partitions. The object of this is to secure the filling of the passages by the stream at three different elevations of the sluice, and thus lessen the loss of efficiency which occurs when the opening of the sluice is too small. E is the disc of the wheel, F is the shaft, G the tail race. KH is a lever which supports the step, and is itself supported by fixed bearings at K, and by the rod L, which can be raised or lowered by a screw so as to adjust the wheel to the proper level. The turbine shown in Fig. 176 is a reaction turbine, but its mechanical construction is the same if it works as an impulse turbine. In the latter case, however, the turbine is not drowned, the discharge taking place above the level of the tail race.

Fig. 177 shows an impulse outward flow turbine. The wheel is of cast iron, and the flow takes place at the opposite ends of a diameter, the two guide passages at either end being made of wrought iron. The valve which regulates the opening of these is made of gun metal, and is keyed to a spindle which can be rotated through the small angle required to close the passages, by hand or by a governor. The following are particulars of two turbines similar to the above:

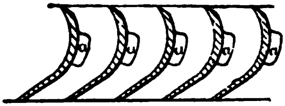


FIG. 178.

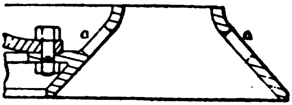


FIG. 179.

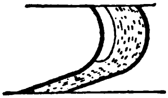


FIG. 180.

	I	II
Inner diameter of wheel	11.81 in.	11.81 in.
Outer diameter of wheel	16.14 in.	14.84 in.
Number of vanes	45 cast iron	84 wrought iron
Revolutions per minute	583	928
Quantity of water per minute	4.52 cu. ft.	5.085 cu. ft.
Head of water	129 ft. 1.6 in.	131 ft. 5.19 in.
Available power per minute	36,440 ft. lbs.	53,806 ft. lbs.
Power measured on brake	16,498 ft. lbs.	28,809 ft. lbs.
Efficiency, per cent	45.27	53.5

Figs. 178 and 179 show sections through the wheel, a a being the ventilating aperatures. The great breadth of the wheel shown in Fig. 179 will be noticed. It is so made to prevent the passages becoming filled at outflow Fig. 180, because if this occurred, the wheel would not work as an impulse turbine should, for the reason that while in reaction turbines, continuity of flow is a necessity, in impulse turbines it must not take place because

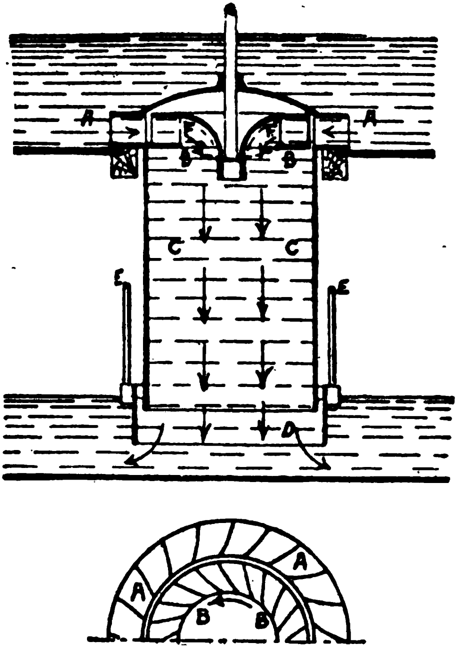


FIG. 181.

the pressure must always be that of the atmosphere, for which reason the ventilating aperatures are provided.

Fig. 181 is an outline drawing of an inward flow turbine with suction tube and cylindrical sluice at the bottom. The lower part of the figure is a sectional plan through guide apparatus and wheel, marked respectively

AA and BB. C is the suction tube, while D is the regulating sluice, by no means an economical method of regulation, because, as the passage at outflow from the sluice is diminished the water flows more slowly through the whole wheel, having also a diminished velocity of whirl at entry, together with a shock. Again, on leaving the wheel it still has some tangential velocity left, because the velocity of the water relative to the wheel is less than when the sluice is raised. The velocity of the stream passing away from D is also increased, thus creating three important causes of loss. The shaft is carried by a collar bearing above. The direction of flow is indicated by the arrows. The letters FF, and the dotted curves below them represent the vanes of an inward and parallel flow wheel, and do not refer to the inward flow turbine. The water is turned from a radial to an axial direction, hence the term "mixed flow" for this type.

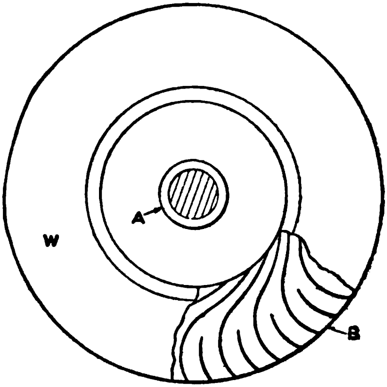


FIG. 182.

The Thompson inward flow turbine is shown in Figs. 182, 183 and 184. Fig. 182 shows the wheel, the portion B being in section, its sides, W, are conical (Fig. 183).

The water enters at O, Fig. 184, and flows around in both directions, as shown by the arrows. Instead of the area being greatest at O and least at the side opposite O, it decreases from O uniformly around the circumference, the water flowing in the direction taken by the hands of a clock, so that no changes in direction may take place.

The guide vanes, D, are hinged at E, and can be moved by links C so that the passage to the wheel is enlarged or diminished, according as the guide vanes move clockwise, or in the opposite direction. This wheel shows good efficiency, either at full load or light load. Alteration of the guide vane angle has been very successful with both inward and axial flow turbines, although some energy is lost by shock at entry with the gate partly closed, and as the volume of water flowing through the wheel is then less, the relative velocity of discharge is less. The water finally leaves the wheel in directions parallel to the axis, and passes out of the casing down the draft tubes, P P.

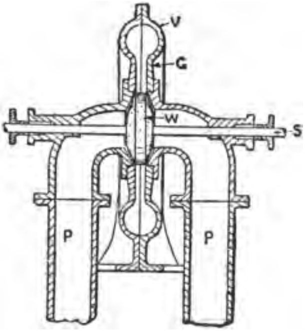


FIG. 183.

The Suction Tube.—As has already been stated, the suction tube enables a reaction turbine to be placed in a higher and consequently more convenient position than it otherwise would be, shortening the shaft and regulating rods, and sometimes, as in the case of the Victor turbine, allowing bevel wheels to be dispensed with, and pivot friction exchanged for journal friction, which latter has been shown by the latest experiments to be less than the former.

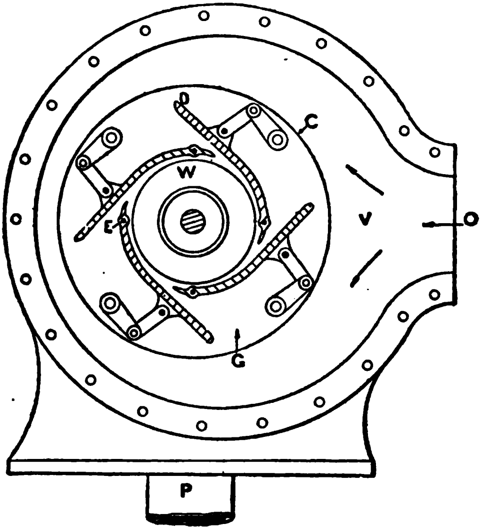


FIG. 184.

Fig. 185 shows a tube, A B, in which is a piston P, C D being the upper level and E F the lower lever of the water. Now, suppose the atmospheric pressure is replaced by the pressure of two columns of water, A C and K B; then the pressure on P is evidently unaltered. But if we express the pressure on P in feet of water, the pressure on its upper surface is A P, while that on its under surface is P K; the effective pressure is therefore A K, and

$$\begin{aligned} AK &= AC + CK \\ &= KB + CK \\ &= BC \end{aligned}$$

so that as long as P is below K where K B is the height of the water barometer, the head is unaltered.

Now, with a given turbine with a given resistance to overcome, and with a given head, no matter how that head is obtained, whether by water pressure or atmospheric pressure, a certain number of revolutions will be obtained, and a certain quantity of water will be used. This is evident, because the state of affairs at the turbine is unchanged, for the pressure at every point is unchanged, as shown above; and if the forces and the sphere in which they act are unchanged, the motions will also be unchanged.

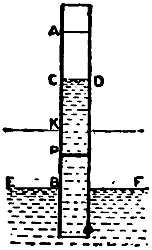


FIG. 185.

This, then, explains how a turbine with a draft or suction tube can be placed above the tail race. The height is theoretically limited to 34 ft., but practically to much less. More accurately, if v_3 be the velocity of flow from the suction tube, theory would give

$$h_3 = 34 - \frac{v_3^2}{2g}$$

where h_3 is the above height.

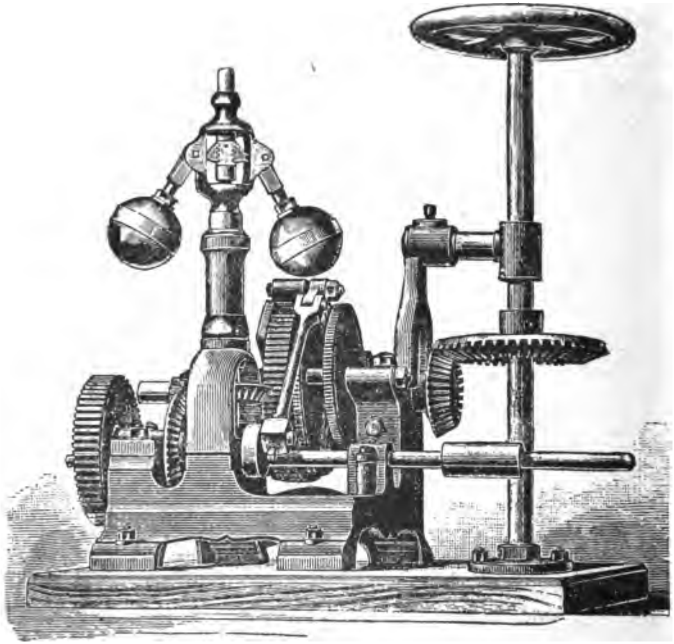


FIG. 186.

In practice, however, in the above equation the following quantities should be substituted instead of 34 for the corresponding diameters of suction tube. This table is given by Meissner.

Diameter of tube in feet.	h_3
.49	31.16
.98	29.52
1.6	27.88
2.3	27.88
3.3	26.24
4.9	19.68
6.5	14.76
8.2	13.77
9.8	12.46
11.5	11.15
13	9.84

Turbine Governors.—All water wheel governors are of the indirect acting class; that is, they enable the wheel to close or open the sluice by its own power, the governor itself not being powerful enough to perform the work directly. Figs. 186 and 187 show two views of the Snow governor as applied to the Victor turbine. Fig. 186 is a perspective view, and Fig. 187 is an outline drawing, showing the principle of its action. Referring to Fig. 186, it will be noticed that there are two shafts, one vertical, and the other horizontal. The former turns the pinion which gears with the spur segment on the sluice, and the latter drives the governor. Upon this horizontal shaft at its right end is a pulley, not shown in the figure, which is driven by belt from the turbine shaft, while at the left end there is a pinion which is concealed by the spur wheel at the extreme left of the figure, which spur wheel the pinion drives, and this, by means of two bevel wheels, drives the governor balls. The arms of the governor have teeth on their inner ends, which are in gear with the central spindle, so that according as the balls fall or rise, the spindle rises or falls. Fig. 187 shows the connection between this spindle and the pawl shifter. B C D is a lever with a fulcrum at C. D G is a link connecting it with the sector, which is called the pawl shifter, and which is pivoted at H. Referring again to Fig. 186, a small crank will be noticed at the right of the governor base, and upon the same shaft that carries the spur wheel at the left of the figure. This crank gives a reciprocating motion to the short connecting rod, at the upper end of which are two pawls, termed the closing and the hoisting pawls, because one closes and the other opens the sluice when they are allowed to gear with the ratchet wheel that lies just at the left of the upper end of the connecting rod. The sector, or pawl shifter (Fig. 187), part of which is just visible to the left of the ratchet wheel in Fig. 186, would, if its upper edge were circular, prevent the pawls gearing with the ratchet wheel; but the depression at the center of the circumference, Fig. 187,

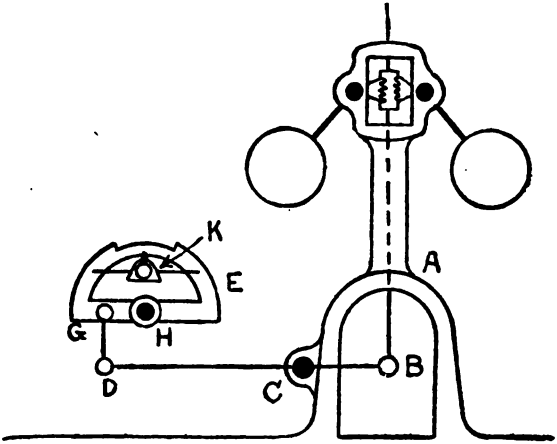


FIG. 187.

enables them to fall into gear when required. For, if the balls rise, B is lowered and D and G are raised, so that the depression on the pawl shifter allows the closing pawl to gear with the ratchet wheel. If the balls fall, motion in a reverse direction takes place, and the hoisting pawl gears with the ratchet wheel. This ratchet wheel turns the vertical shaft by means of the two bevel

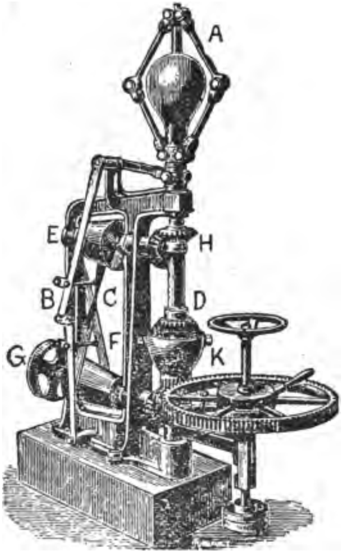


FIG. 188.

wheels shown at the right of Fig. 186, the larger of which is on the vertical shaft. In order to prevent overwinding, there is an arrangement of reducing gear between the ratchet wheel and bevel pinion, which brings into position a stop similar to the sector, or pawl shifter, which will allow the pawls to slide freely along its top, and prevents them engaging with the ratchet wheel, and thus prevents further motion of the sluice. In order to regulate the speed, there is a horizontal bar on the pawl shifter, upon which slides a weight, K, Fig. 187. If this is pushed to the right, it will tend to raise D and lower B, and lift the governor balls, so that in order that the pawl shifter may be in its middle position, with neither pawl in gear, the speed of the wheel must be less. For a similar reason, if is desired to increase the speed, K should be shifted to the left. This governor is said to be very rapid in its action. Hett's governor is shown in Fig. 188. Its action depends mainly upon the differential motion of three bevel wheels.

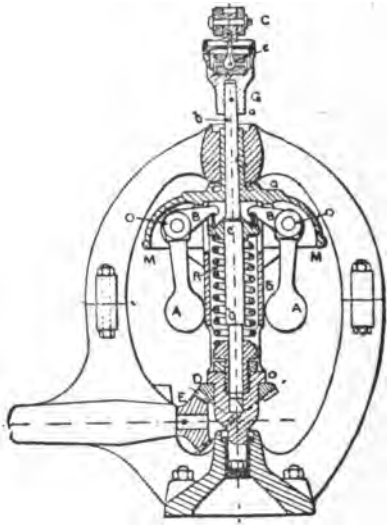


FIG. 189.

In this governor the initiative is given by a governor of the Porter type A, fitted with heavier balls than usual. As this rises or falls it moves a belt shifter, B, which changes the position of the cross belt C on the cone pulleys E and F. Thus when the governor sleeve is in mid position, the belt is on the center of the cones and E is driven at the same speed, but in the opposite direction to F, which receives its motion from the pulley G. When

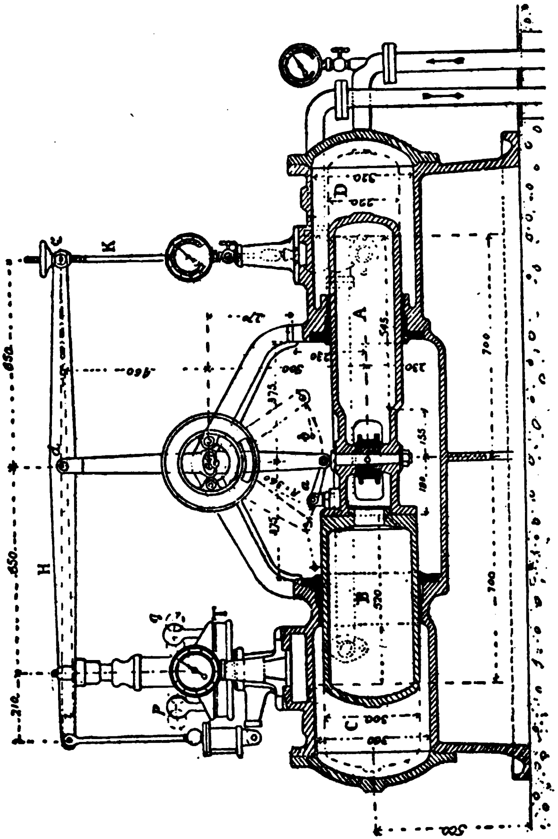


FIG. 190.

the sleeve falls the belt is shifted to the left, so that E is driven faster than F, and when the sleeve rises the reverse takes place. The lower cone shaft drives the governor spindle by means of miter bevel wheels. The upper cone shaft drives a sleeve H D, which is loose on the governor spindle, and carries the upper bevel wheel of differential motion D at its lower end. The lower pinion of D, being attached to the governor spindle, revolves with it. As the gearing is the same, it is clear that when the governor is running at its proper speed, so that the belt is on the middle of the cones, the upper and lower bevel wheels of the differential motion run at the same speed in opposite directions, and the two other bevel wheels in the cup K rotate on their axes, but do not give motion to cup K. When the upper bevel wheel runs at a different speed to that of the lower one, the cup K rotates, and moves the pinion at its base, which gears with the spur wheel on the hand-wheel spindle of the starting gear. To connect the motion of the hand-wheel spindle, a spring clutch is employed, which can

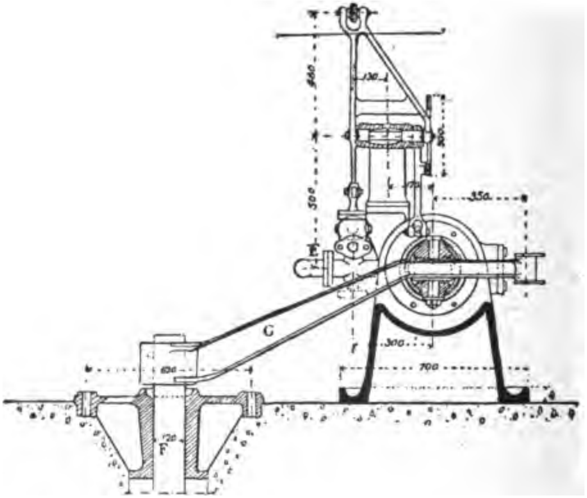


FIG. 191.

be instantly thrown in or out of gear by giving a quarter of a turn to the disc, shown immediately above the spur wheel. The clutches are thrown out when stopping by hand, and will slip if the running speed is not reached when the gate is fully opened. One great advantage of this arrangement is that the speed of regulation increases in proportion to the amount of irregularity to be corrected.

Fig. 189 shows a form of governor constructed by the Vevey Iron Works. The balls, A A, are carried by the bell cranks, B B, having knife edges, O O, which are carried by arms, M M, that are keyed to the governor shaft, a a. The ends, B B, of the bell cranks rest upon a sleeve, c, which carries the movable central rod, b b. The small pieces of steel between B B and the sleeve should be noticed. They reduce the friction considerably, as they are free to turn through a small angle.

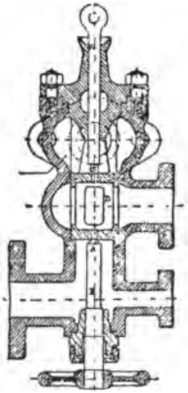


FIG. 192.

The part C acts on the valve of the servo-motor, whose duty it is to move the sluice or gate of the turbine. The servo-motor of the Bellegarde turbines is illustrated in

Figs. 190 and 191. The water under pressure acts upon a differential piston, A B, connected at the center to a lever, G, which turns the shaft, F, controlling the sluice. D is permanently connected to water under pressure, while C is put in connection by the governor, T, with water under pressure, or with the discharge. In the first case the differential piston moves to the right, and in the second case it moves to the left. The former motion closes, and the latter opens the sluice. The distribution of water under pressure to C is effected by a valve S, shown in detail in Fig. 192. This valve rests on the upper orifice of cylinder D, while the other orifice at the bottom is throttled more or less by the point of the regulating needle, A B, Fig. 192. The valve S is suspended by the rod K at the point e of the lever H, which is pivoted at d, and supported by the index i of the governor, T. When the governor moves i, it produces at the same time a motion of valve S. As the lower orifice of cylinder D, Fig. 192, is connected to the water under pressure, while the upper orifice is in connection with the discharge to the atmosphere, and the space between these two orifices is in communication with the large cylinder, C, it will be seen that when the index of the governor rises, and the valve S is lowered, the upper orifice is closed, and water under pressure flows into cylinder C through the lower orifice. When, on the other hand, the governor index descends, valve S rises and opens fully the upper orifice, which has a section almost double that of the lower, so that the pressure falls in C and the water escapes from it. If, now, the valve is midway between these two positions, so that the two orifices present the same section to the water, the pressure in the cylinder C remains about midway between the initial pressure of the water and the pressures on the differential piston A B are balanced so that it does not move. There is, however, this disadvantage, that there is an

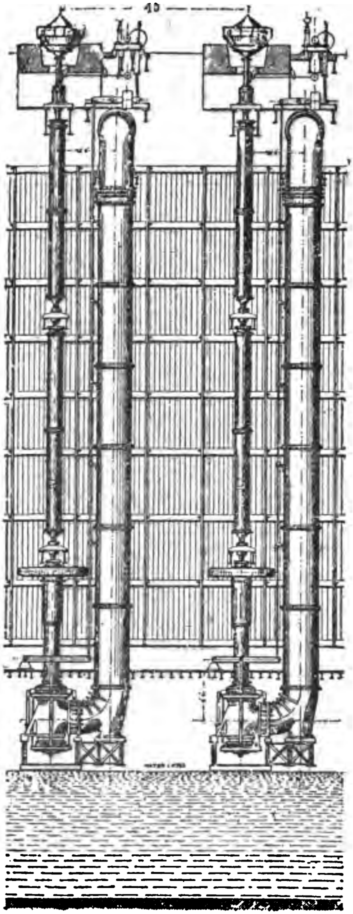


FIG. 193.

almost continual flow of water through m. On the other hand, there are two great advantages. First, the small valve S is only rarely forced to the end of its stroke by the governor. Generally, it is only partially moved,

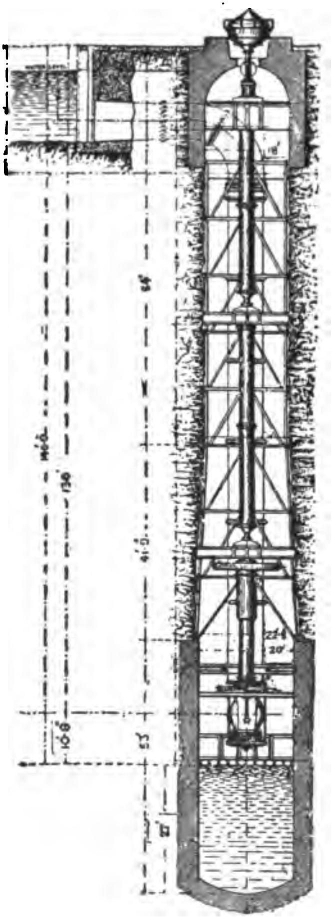


FIG. 194.

and from this results a speed of gate opening, or closing more or less great.

It is only when there is a considerable change of load that the valve is pushed down by the governor, and the velocity of the motion of the piston, A B, reaches its maximum value. A small change of load, on the other hand, produces a change in the opening and closing of the sluice proportional to the magnitude of the change of load. The small valve is acted on by the liquid with a variable force. When it descends, the upward force on it increases from zero to several pounds, and this gives the governor stability. The pistons must be so proportioned that the resistance of the sluice is easily overcome, and the orifices in D, Fig. 192, must also be of such a size that the rapidity with which the sluice is closed may be such as is desirable. Generally, orifices of from 0.24 in. to .39 in. will suffice.

So far, apparently, the governor only puts in and out of gear the apparatus for closing the sluice, and for that purpose only a very small motion of the index is necessary; but in practice the piston, A B, and consequently the sluice, takes up, when the wheel is running steadily and the load is constant, a position determined by that of the index of the governor. The point of rotation d of the lever H is not really fixed; it is connected to the piston, A B, in such a manner that it rises or falls proportionally to the motion of this piston. It results from this, that when the piston is at rest, and consequently the valve S occupies its mean position as well as the point e, the index i of the governor is obliged to fix itself in the position corresponding to that of the point d, and, reciprocally, the point d and the piston, A B, are obliged to follow the movement of the index i. But when the wheel speeds are in a state of oscillation the index i acts freely upon e independently of the motion of d. R garding the motions of d and A B, the piston, A B, moves by the connecting rod a b the crank b c, which carries at C an eccentric, whose rotation raises or lowers the point d by means of the rod d c. The displacement of d is approximately proportional to the movement of A B.

Turbines in Use at Niagara Falls.—Figs. 193, 194 and 195 show the general arrangement of these turbines, which were designed by Foesch and Piccard of Geneva, and built by I. P. Morris & Company of Philadelphia. Each turbine operates under an average head of 136 feet, and consists of two wheels to each shaft.

The speed is 250 R.P.M., with a volume of water equal to 430 cubic feet per second, and each turbine develops 5,000 horsepower. The shaft is vertical and drives an electric generator direct connected at the top. The wheels are of bronze, and are shown in Figs. 196 and 197, while the lower wheel is shown to a larger scale in Figs. 198 and 199. They are reaction wheels, although the fact that they discharge into the atmosphere might lead one to suppose at first that they were impulse turbines.

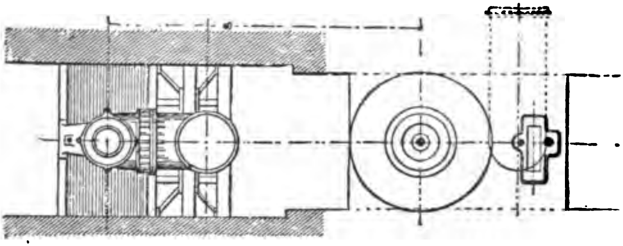


FIG. 195.

There are 36 guide, and 32 wheel vanes. A steel pipe 90 inches in diameter conducts the water from the canal (Fig. 200) to a chamber between the two guide wheels.

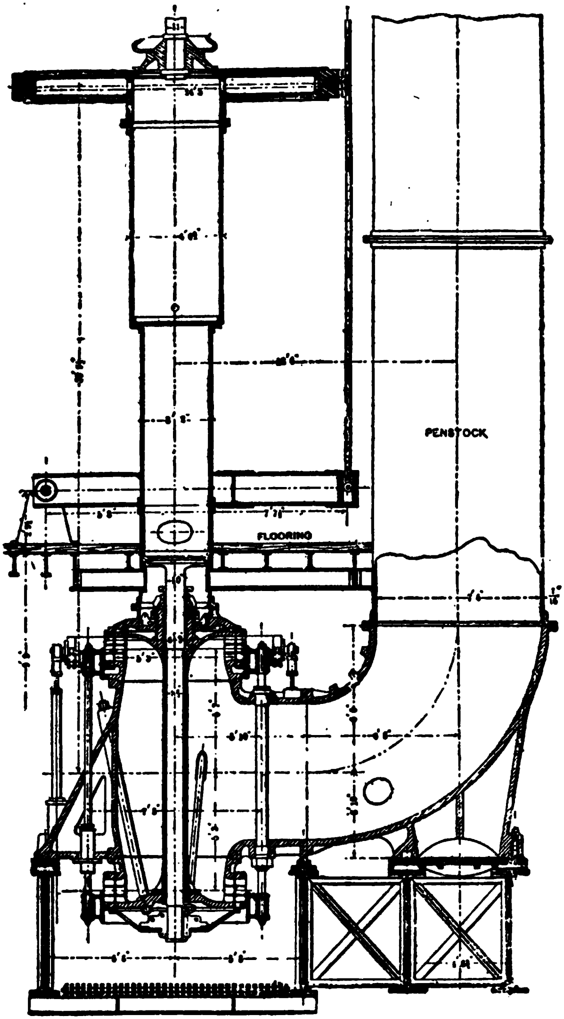


FIG. 196.

The main shaft is made of steel tubes 38 inches in diameter, and 11-inch solid shafts are interposed at the points where the guide bearings are placed. This method of construction at once lightens the weight, and increases the rigidity of the shaft, and also reduces the number of guides necessary. A heavy fly-wheel of forged iron, weighing 10 tons, and having a diameter of 14 ft. 6 in. is attached to the shaft, as shown in Fig. 196. The peripheral speed of this wheel is 11,000 ft. per minute. The upper bearing of the shaft is entirely relieved of its load when the turbine is running, due to the pressure of the water acting upon the under surface of the upper wheel. The speed is regulated by ring sluices, not between the wheel and guides, but outside the wheel.

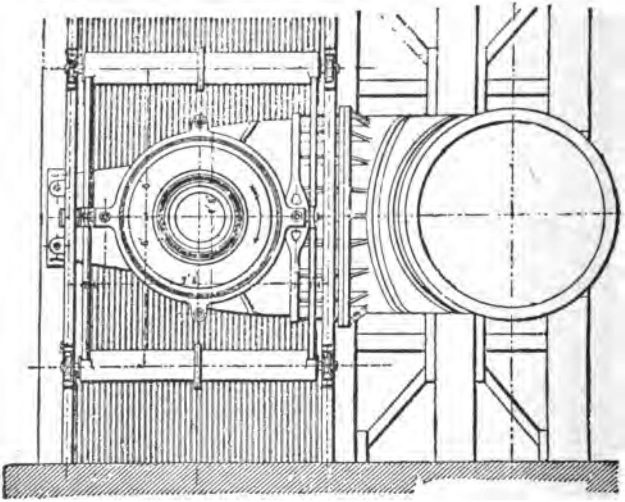


FIG. 197.

The sluice is raised or lowered by means of a system of rods and levers (clearly shown in the figures), which are under the control of a very sensitive governor of the indirect acting class. The speed varies between a maximum of ½ per cent at the normal rate of working, and about 3 per cent if the load is increased or diminished suddenly by one quarter. This sensitiveness is very necessary for an electric installation.

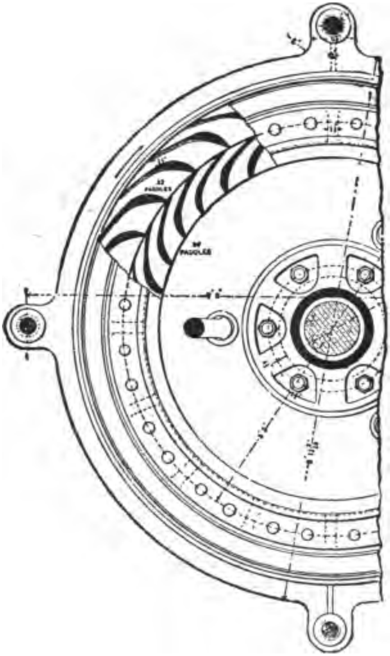


FIG. 198.

A tunnel has been made to carry off the water from the wheels. It is 6,700 ft. long, with a section of 490 square feet, and a hydraulic gradient of about 7 ft. in 1,000, and discharges its water into the river again below the Falls, and near the Suspension Bridge. The velocity of flow is given by Professor Forbes as 25 ft. per second, which we believe to be unprecedented. Ordinarily, about 8 ft. per second is considered as the extreme limit in brick-lined sewers; indeed, a leading waterworks engineer allows only about 3 ft. as a maximum in lined channels. The lining here is made of 16 in. of hard brick, this having been found necessary because the rock passed through disintegrated or crumbled quickly when exposed to the air. The brick and hydraulic cement used in the construction of the lining had united to form a mass as solid as rock in some specimens taken from the tunnel.

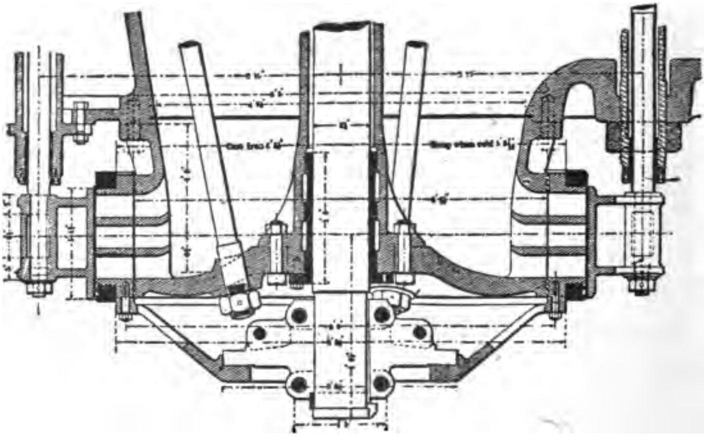


FIG. 199.

The "Poncelet" Water Wheel.—Fig. 201 shows a type of water wheel known as the "Poncelet" that is adapted to falls not greater than 5½ feet. The stream flows under a sluice, and enters the buckets of the wheel without shock, owing to the curvature of the vanes. The stream then rises up in the vane on account of its relative velocity, v, and falls again, flowing out radially. The sluice is inclined at an angle of about 50 degrees. Some of the head must be lost if the wheel does not work in back water, as Fig. 201 shows the water leaving the wheel at some little height above the tail race. Theoretically, the arrangement should be as shown in Fig. 202, where D E is a horizontal line and the angle D E C is bisected by the line C A, so that the fall of the water, after it has attained its greatest height in the wheel is the same as its rise.

Neglecting friction, $v_2 = v_1$, $\theta = \eta - \Phi$, and $c_1 = c_2$.

$$\left.\begin{aligned} c_1 &= v_2 \cos \theta \\ \text{and} \qquad \frac{c_1}{\sin(\alpha + \Phi)} &= \frac{v_1}{\sin \alpha} \end{aligned}\right\} \ldots\ldots\ldots (1)$$

$$\sin(\alpha + \Phi) = \sin \alpha \cos \theta = -\sin \alpha \cos \Phi \ldots\ldots (2)$$

$$\left.\begin{aligned} 2 \sin \alpha \cos \Phi &= -\cos \alpha \sin \Phi \\ 2 \tan \alpha &= -\tan \Phi \\ \text{Generally,} \qquad D\,C\,A &= \text{about } 15^\circ. \end{aligned}\right\} \ldots\ldots\ldots (3)$$

$$\alpha = 15^\circ \ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

An efficiency of about 70 per cent has been obtained with this wheel, but this is rather high for calculation, 66 per cent being more suitable.

The Pelton Water Wheel.—This wheel, also known as the tangential water wheel is shown in Fig. 203. It is made up of a number of double buckets, fixed to the circumference of a wheel. Water projected from a nozzle strikes the buckets at their centers, and is deflected equally to both sides.

Let v = velocity of water projected from nozzle
c = velocity of center of buckets
v_1 = relative velocity of inflow
v_2 = velocity of outflow.

A section of a bucket is shown in Fig. 204, and it will be seen that shock will occur at inflow owing to shape of the central partition, which theoretically should be like a knife edge, but as this is impracticable, it will be ignored for the present, and assuming that the coefficient of resistance referred to v_2 is F, the following equations appear:

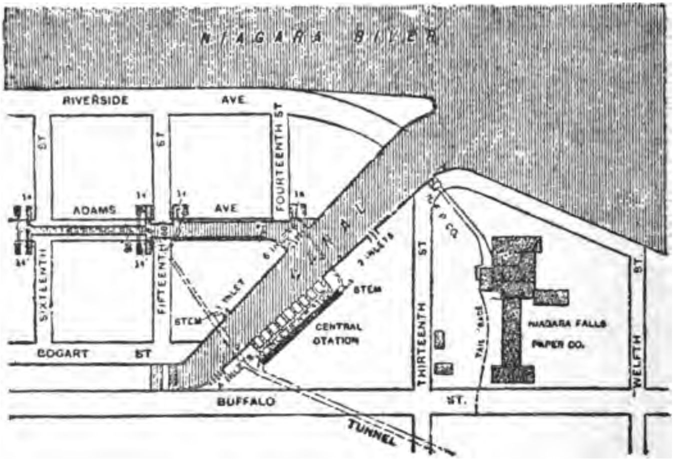


FIG. 200.
CANAL—NIAGARA FALLS POWER STATION.

$$\text{Work done by wheel} = \frac{c\,(w_1 - w_2}{g},$$

in which w_1 and w_2 are the tangential velocities of the water at inflow, and outflow,

and

$$w_1 = v$$
$$w_2 = c - v_2$$
$$v_1 = v - c$$

$$\frac{v_2^2}{2g}(1+F) = \frac{v_1^2}{2g} = \frac{(v-c)^2}{2g}$$

The total wasted energy is:

$$\frac{Fv_2^2 + W_2^2}{2g},$$

which must be a minimum for maximum efficiency. We therefore have

$$(1)\ \frac{F}{I+F}(v-c)^2 + (c-v_2)^2 \text{ as a minimum}$$

$$= \frac{F}{I+F}(v^2 + c^2 - 2\,vc) + c^2 + \frac{(v-c)^2}{I+F}$$

$$-\frac{2c\ (v-c)}{\sqrt{1+F}}$$

$$(2)\ \frac{F}{1+F}(2c-2v) + 2c - \frac{2\ (v-c)}{1+F}$$

$$-\frac{2v-c}{\sqrt{I+F}} + \frac{2c}{\sqrt{I+F}} = 0$$

(3) $c = \frac{v}{2}$ whatever the value of F.

Figs. 205 and 206 show improved types of the Pelton wheel, in which the streams of water strike the buckets alternately on each side of the continuous dividing edge as shown in the cuts, thus securing a steadier motion. The impulses are divided more regularly on the wheel, as each bucket passes the point of the nozzle and

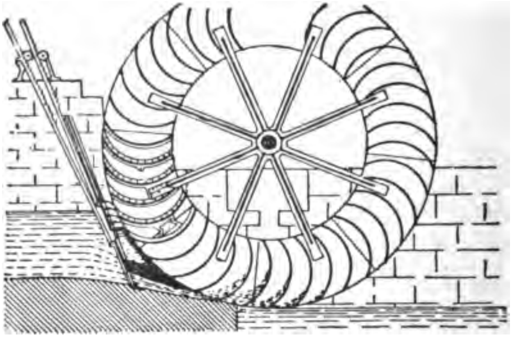


FIG. 201.

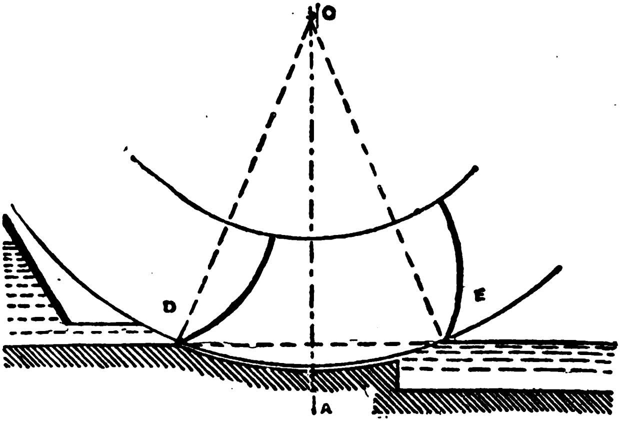


FIG. 202.

catches its share of the water. These buckets are cast solidly upon each side of the circular dividing ridge, and upon the face, or rim of the wheel on each side of this central division. This circular ridge being also angular, and curved as it approaches the center, gives to the interior of the buckets a symmetrical, and effective curve.

The further claim is made by the builders that this arrangement of buckets, and form of construction secure great strength, rigidity and stability, and the buckets are not subject to the difficulty of becoming loose. In this type of wheel several jets are used. The method of mounting is such as to permit the moving of these jets, and thereby varying the angle of inclination at which the water may be projected on the buckets. Either of the nozzles may be removed, and others of different size, or bore substituted, or any of them may be capped over, and one or several, or all used as may be desired.

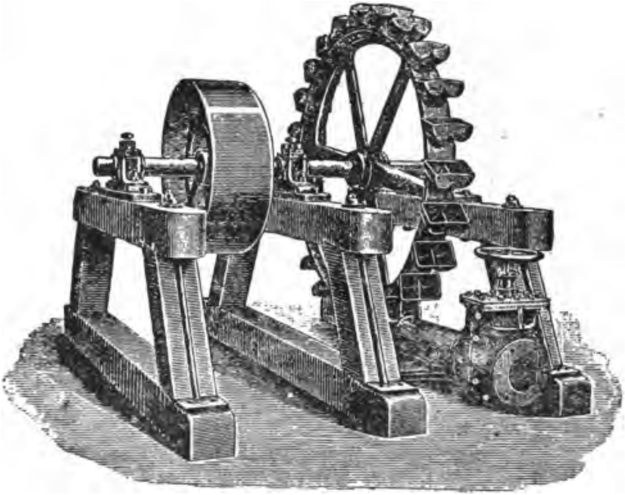


FIG. 203.

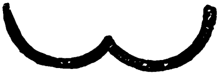


FIG. 204.

SETTING HYDRAULIC RAMS.

Hydraulic rams are water-raising appliances in a class by themselves. The shock that is commonly noticed on quickly closing a fullway cock and suddenly stopping the flow of water in long lengths of pipe is the power employed in the hydraulic ram.

FIG. 205.

The hydraulic ram is a machine of very simple construction, but it does not seem to be understood as it should be by those who may have the erection and installing of it. It consists of a body, or trunk, and an air-vessel, in which are inserted three valves, namely: (1) The pulse, or foot valve at end of the trunk; (2) the retaining, or ascension, valve, in the air-vessel; and (3) the snifting valve, in the neck of the air-vessel, immediately under the ascension valve. The office of the snifting valve is to supply air to the air-vessel when the ram

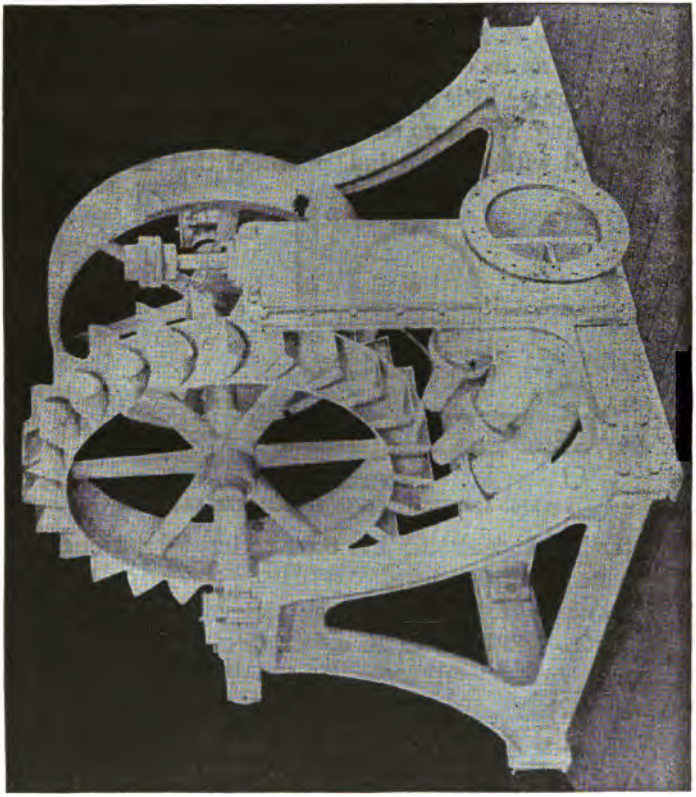


FIG. 206.

is working. This, though seemingly the most unimportant valve in the ram, is most necessary to the successful working of the machine, as without it the ram would soon cease working, owing to the air having been exhausted, as the air-vessel would have become waterlogged. Air escaping from the air-vessel with the water causes the severe shocks and noise in badly constructed rams that are not fitted with this simple valve.

The hydraulic ram is a machine that utilizes the momentum of a stream of water having a slight fall, in such a way as to elevate a portion of that water to a greater height. This machine is self-acting; and when once set going, it will go on working day and night for a long period without stopping, provided that the supply of water is sufficient and that the ram has been properly constructed, and fitted up.

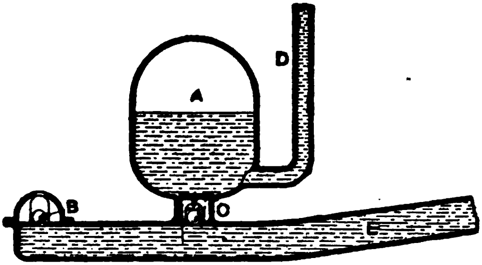


FIG. 207.
SECTIONAL ELEVATION OF HYDRAULIC RAM.

The hydraulic ram will force water with a fall of from 18 in. to 100 ft. to an elevation of 15 ft. to 1,000 ft. from almost any distance, from a few score of yards to four or five miles, or more if necessary. For example, assuming that 100 gal. of water falling 10 ft. would elevate 10 gal. to a height of say 80 ft., as 100 gal. falling 5 ft. will elevate 1 gal. to a height of 300 ft., a hydraulic ram will raise water from 300 gal. up to 500,000 gal. per day of twenty-four hours if required.

The following example will give some idea of the power exerted at the end of a long pipe when the flow is suddenly stopped: A pipe flowing full bore with a velocity of 25 ft. per second is equal to a head of 10 ft. If the pipe is 2 in. diameter and 150 ft. long, the contents will be $\frac{2^2 \times 62.5}{144} \times .7854 \times 150 = 204.5$ lbs., which multiplied by 10 = 2,045 ft.-lbs. of energy.

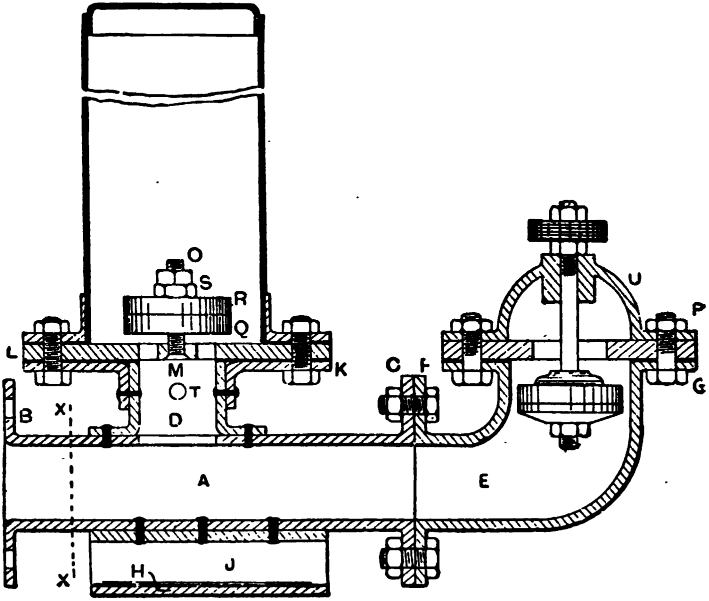


FIG. 208.
VERTICAL SECTION OF COPPER HYDRAULIC RAM.

A diagrammatic section of a hydraulic ram is presented by Fig. 207. A is an air vessel, B and C ball valves, D the delivery pipe, and E the supply pipe. Above valve B is an opening, and the water running down from a small fall at E, passes this outlet until the velocity is sufficient to close B. This of course suddenly stops the stream, and the outlet valve is forced open, owing to the great increase of pressure in the ram. Through C the water passes into A and up the delivery pipe D. This releases the pressure, and valves B and C drop to their seats, and the operation is repeated. In some cases an ordinary lift, or flap valve, weighted to slightly exceed the static pressure of the supply stream, is placed between E and C. Obviously, a small portion only of the supply water from a small fall is delivered to a greater height, and the average efficiency of the ram is probably not more than 50 per cent.

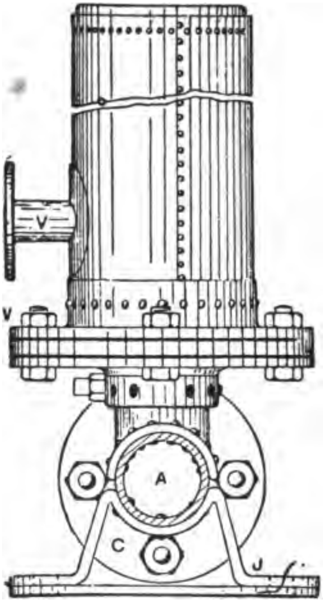


FIG. 209.
END ELEVATION OF COPPER HYDRAULIC RAM.

Figs. 208 and 209 show sectional, and end elevations of a 1-inch ram in copper sheet or tube adapted for springs or streams supplying from 90 gals., to 300 gals. per hour, and raising approximately 5 to 20 gals. per hour to a height of 80 feet. These illustrations are so proportioned however, that a ram may be constructed from them to suit any fall of the drive pipe, and any quantity of water, by simply increasing the dimensions according to the varying conditions of the stream, or head.

The volume of water raised by a hydraulic ram varies according to the ratio of the fall to the height that the water is to be lifted, while the effective capacity is materially affected by the length of the drive and delivery pipes, as well as by the relative fall and lift. A ram will work on a fall as low as 2 ft. 6 in., and the quantity of water raised will be in proportion to the fall. Generally speaking, a proportion of 1 in 10 of fall to lift gives the greatest efficiency. As there are no two streams whose conditions are exactly alike, it is obvious that a ram which will work well on one stream, will not work satisfactorily on another. Therefore, to obtain the highest efficiency, the ram must be designed to suit the conditions of the stream on which it is to work. These conditions are ascertained by surveying the site, and mak-

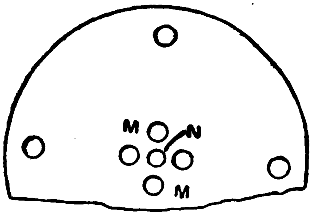


FIG. 210.
PLAN OF DELIVERY VALVE SEAT FOR HYDRAULIC RAM.

ing other observations. The data required are as follows: (1) the minimum volume of flow per minute supplied by the stream in summer; (2), the horizontal distance from the stream to the proposed site of the ram; (3), the vertical height that the water is to be lifted above the ram; (4), the vertical height obtainable from the source of supply; (5), the horizontal distance from the ram to the end of the delivery pipe. The method of making these observations will be described later on. The ram illustrated in Fig. 208 and 209, may be constructed of sheet-copper, or drawn tube, the latter being the most satisfactory. In order to withstand the pressure, the thickness should be not less than 3/32 in. for low falls of 5 feet, and a short drive pipe.

For falls above this height, and for long drive pipes, the thickness should be 1/8 in. Fig. 208 is a section of the ram, and Fig. 209, an end elevation, with the connecting flange cut away on line xx (Fig. 208). The body A is 5 3/8 in. long when finished, and to allow for the flanges B and C, the sheet or tube should be 7 1/8 in. in length. The width of the sheet (if used) will need to be sufficient to make a tube of 1-in. internal diameter, and to allow for a 3/8-in. lap for the joint. Small rivets are used for making the joint, which is afterwards soldered. The edge of the lap on the inside should be bevelled off, and care should be taken that the heads of the rivets do not project much into the ram. The flanges are worked over as shown, to 2 3/4 in. in diameter. At 1 3/4 in. from one end a hole 1 in. in diameter is cut, the center of the hole being 2 1/4 in. from the end; over this hole is riveted the T-piece D, communicating with the delivery valve.

The T-piece is of the same diameter as the body of the ram, and when finished it is 1 inch high. The sheet or tube will need to be cut sufficiently long to allow of a flange ¾ in. wide, being worked on the end resting on the body of the ram. The T-piece is riveted to the body as shown, and it is also soldered so as to make a water tight joint.

To avoid having rivet heads on the inner surface of the ram, the T-piece and body are bolted together with

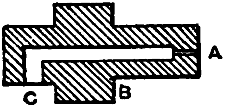


FIG. 211.

SECTION OF SNIFTING VALVE FOR HYDRAULIC RAM.

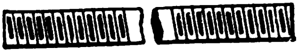


FIG. 212.

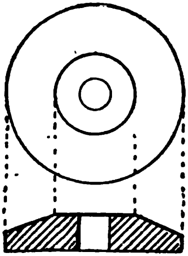


FIG. 213.

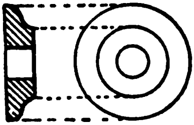


FIG. 214.

DETAILS OF PULSE VALVE OF HYDRAULIC RAM.

⅛ inch brass wire, having a fine thread cut on it, the holes in the flange of the T-piece and body being tapped for this thread. When all the studs are screwed up, they are cut off nearly flush, say to about 1/64 in. of the inside and outside surface of the copper, by filing, after which a 1-inch iron bar is inserted in the tube, and the brass studs are riveted down by a few blows on the outside with a hammer. The joint is then sweated together with solder, and if there are any burrs left on the inside ends of the studs, they should be smoothed off with a file, to prevent obstruction to the passage of the water. The elbow E, Fig. 208, is of copper tube 1 inch internal diameter. For rams working on even low falls the thickness should be not less than 1⅛ in. so that the bend may be worked out to the larger diameter as shown, the elbow being enlarged where the pulse valve works to an internal diameter of 1⅝ in. The tube should be of sufficient length to allow of the flanges F and G being worked over.

Flange F which is bolted to the body, is 2¾ in. in diameter, and flange G is 3⅜ in. in diameter. The approximate length of tube required is from 7½ in. to 8 in. Holes for ¼ in. bolts are drilled through the flanges as shown. The bed plate H and J, Fig. 208, is made of two pieces of sheet brass ⅛ in. thick. The bottom piece H is 3¼ x 4 in., and to this is bolted and sweated, the cradle J, Fig. 209, made out of a sheet about 3¼ x 6¾ in., shaped as shown, to fit the body of the ram.

The bed plate and body are riveted or bolted, and sweated in the same manner as is the T-piece. The flange K, Fig. 208, is ⅛ in. in thickness by 4 inches in diameter. From the center the metal is worked out at right angles to form a ring with an internal diameter of 1¼ in., this ring fitting over the T-piece, to which it is secured by riveting.

To this flange is bolted the circular valve seat L, a plan of which is shown in Fig. 210. This valve seat is made of a brass plate 4 in. in diameter by ¼ in. thick. The four water ways M, M, are ¼ in. in diameter, and the center hole N is drilled and tapped for a ¼-in. stud O, Fig. 208, on which the delivery valve works. The stud after being screwed in, is riveted over on the under side as shown.

The water passages are drilled with their centers 3/8 in. from the center of the stud. The pulse valve seat P is a brass plate 3 3/8 in. in diameter by 1/4 in. thick. A water way 1 in. in diameter is drilled in the center, this

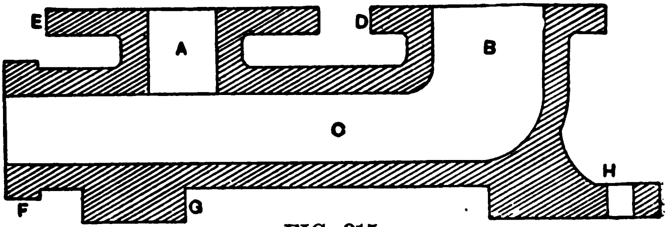


FIG. 215.

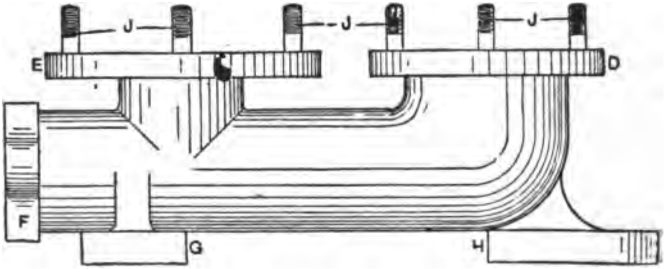


FIG. 216.

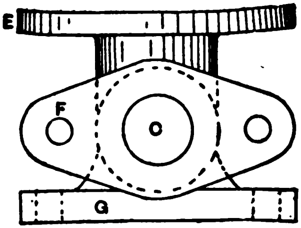


FIG. 217.

SECTION, SIDE, AND END ELEVATIONS OF CAST-IRON HYDRAULIC RAM BODY.

being the diameter of the bore of the ram. Four holes for ¼-in. bolts are drilled in both valve seats. For larger rams the number and size of the bolts should be increased, the spacing being 2 in. from center to center. The delivery valve consists of a rubber disc Q, and a brass disc R, both being 1⅜ in. in diameter by ¼ in. thick.

Through the center of each, a hole is made for the stud O, which should fit loosely. The brass disc acts as a weight to keep the rubber disc in position, and the lift of the valve is adjusted by the nuts S. The delivery valves of small rams should not have more than ⅛-in. lift. For larger sizes the lift is proportionately greater.

A small split pin is put through the stud above the nuts, to prevent their working loose. The snifting valve or air inlet, is shown at T, Fig. 208, and in section in Fig. 211. It consists of a brass plug having a ⅛-in. hole drilled through its center, tapering to a very fine hole at the inner end A, where it should not be larger than would allow a very fine needle to pass through it. The enlarged part B is hexagonal for the wrench, and at C, a hole is drilled at right angles to the first, to guide the water downward, and prevent splashing, as a small quantity will squirt out, when the pulse valve closes. The snifting valve is screwed into the T-piece through the lap of the flange K (see Fig. 209) in place of one of the rivets. The object of this valve is to allow a little air to pass into the ram at each beat of the air vessel.

The details of the pulse valve are shown separately at Figs. 212, 213 and 214. The spindle (Fig. 212) of the valve is a ¼-in. diameter mild steel rod 3½ in. long with a thread cut at each end for a length of 1¼ in. A brass disc (Fig. 213), 1⅜ in. in diameter by ¼ in. thick, tapered off at its outer edge as shown, forms the under support of a rubber disc, 1⅜ in. in diameter by ¼ in. thick. The upper brass disc (Fig. 214) is ⅞ in. in diameter by ¼ in. thick. The central hole in the brass disc is tapped to fit the thread of the valve spindle.

In putting the valve together the upper brass disc is first screwed up tight on the spindle with the flat side downwards (see Fig. 208); over this is placed the rubber disc; and over this is screwed the supporting brass disc, which must not be screwed up so tight as to press the rubber out of shape. Below this disc is a lock-nut; and further to prevent the disc and nut working loose while the ram is working, a small hole is drilled through the spindle for a small split pin.

At the other end of the spindle are two nuts and brass washers for adjusting length of stroke, and weight of the pulse valve according to the length and fall of the drive pipe, and the height to which the water is to be raised. The guide sleeve U for the pulse valve spindle is made of a strip of sheet brass ¾ in. wide by ¼ in. thick, bent as shown, the central portion being ¾ in. long and ½ in. in diameter. This sleeve must be fixed perfectly square with the lugs to allow the valve to rest squarely on its seat. The valves are faced with rubber in order to reduce noise and wear.

The air chamber, Fig. 209, has an internal diameter of 2¼ in. and is made of sheet copper 1/16 in. thick, riveted and soldered together. The flanged copper pipe v near the bottom is the outlet to the discharge pipe, the diameter depending upon the length of the delivery pipe. With a long delivery pipe, the diameter for this sized ram should be not less than ½ in., but with a short delivery ⅜ in. would be sufficient.

It is unnecessary to go into details regarding the construction of this portion of the ram, as the cut explains itself. Fig. 215 is a sectional elevation of the body in cast iron. Fig. 216 shows a side elevation, and Fig. 217 an end elevation of the body. A is the water way to the delivery valve and air chamber. B is the passage leading to the pulse valve, C waterway in the body, and D flange to which pulse valve seat and guide valve are bolted. E is the flange to which are bolted the delivery valve seat, and air chamber, F flange for connecting drive pipe, and G back lugs for bolting ram to the foundation, there being a bolt on each side of the ram. H is a lug cast at the end of the elbow for bolting the front to the foundation, and J short studs ¼ in. in diameter used for bolting the valve seats and the other fittings to the ram. In order to obtain the highest degree of efficiency from a hydraulic ram water-raising plant, the horizontal length of the drive pipe should equal the vertical height that the water is to be raised, and the diameter should be such as to discharge three times the available quantity. The diameter of the delivery pipe should be such as not to add more pressure on the ram than is due to a head of 3 feet.

The area of the pulse valve water way should equal that of the drive pipe, and the area of the delivery valve water way should be as large as possible, to permit a free passage for the water, with a short lift of the valve, thus preventing the water flowing back past the valve, causing slip. The capacity of the air chamber in cu. in. should not be less than the cubic capacity of the delivery pipe.

Flow of Water.—When gauging streams with a very small quantity of water flowing, a triangular notched weir should be used, as this gives more accurate results with small quantities than a rectangular weir.

The formulæ for making hydraulic ram calculations are:

$$(1)\quad g = \frac{G \times f}{L} \times 0.65$$

$$(2)\quad G = \frac{g \times L}{f} \times \frac{1}{0.65}$$

$$(3)\quad f = \frac{g \times L}{G} \times \frac{1}{0.65}$$

where G = gallons of water flowing down the stream to work the ram; g = gallons of water raised; L = lift, or height the water is to be raised above the ram; f = the vertical fall of the feed water; and 0.65 the approximate efficiency of the ram described in this article.

To show the working of these formulæ, a stream is taken as an example, having an available fall of 5 ft., with 2.819 gal. flowing per minute. It is required to raise the water to a tank 50 ft. above the ram, the length of the delivery pipe being 60 ft. What quantity of water would be raised to the tank in 24 hours? Applying formula 1, the quantity raised, assuming that there is practically no frictional loss in the delivery pipe, will be $\frac{2.819 \times 5}{50} \times 0.65 = .1832$ gal. per minute, or $.1832 \times 60 \times 24 = 263.8$ gal., say 260 gal. in 24 hours, with 4,060 gal. passing through the ram in 24 hours. Now, applying formula 2, what quantity of water will be required to raise .1832 gal. to a height of 50 ft. above the ram, the supply water having a fall of 5 ft.? The quantity will be $\frac{.1832 \times 50}{.5} \times \frac{1}{0.65} = 2.819$ gal. per minute.

The working of these two examples will show the use of the formulæ, and from the data obtained by them, the dimensions of any ram may be calculated.

In calculating the required diameter of a ram to pass a given quantity of water through a given length of drive pipe, on a given fall, the quantity that is to pass through the ram is multiplied by 3, for the reason that the water in the drive pipe flows only about 1/3 of the time towards the ram, and the diameter must be calculated for the maximum flow at any moment.

Taking the above quantity of 2.819 gals. per minute on a fall of 5 feet, and assuming that the water is to be raised 50 feet above the ram, the length of the drive pipe will be $50 \div 3 = 16\ 2/3$ yards, or say 17 yds. long, and the maximum flow will be $2.819 \times 3 = 8.457$ gals. The required diameter of the ram, will by this rule, be $\sqrt[5]{\frac{8.457^2 \times 17}{5}} \div 3 = 1$ in. in diameter. The maximum quantity that would pass through a ram of a given diameter, length of drive pipe, and fall is found by the rule for ascertaining the discharge of a pipe in gals., and dividing the result by 3. The rule follows: multiply area of pipe in square inches by velocity of water in feet per minute and divide the product by 144; the result is number of cu. feet discharged per minute. Multiplying this by 62.5 will give number of pounds, which divided by 7.5, will give number of gals.

The diameter of the delivery pipe for a given frictional loss is found by the same rule as that for the diameter of the ram; but instead of dividing by the available head, the head to be lost in friction is taken. In this case the quantity flowing through the pipe is not multiplied by 3, as the water has a practically constant flow.

For short delivery pipes the head due to friction may be ignored, but for lengths above 150 feet the head that will be lost must be added to the actual height the water is to be raised, and the diameter must be calculated for this loss. The capacity of the delivery pipe is found by multiplying the area of the pipe in sq. inches by its length in inches. The result will be the capacity of the pipe in cubic inches, and the air vessel will have to be of the same capacity. In the case of long delivery pipes, to prevent the air vessel being of inconvenient length, it should be made baloon-shaped. By these simple rules it is possible to calculate the dimensions of a ram that would work satisfactorily with water from any stream.

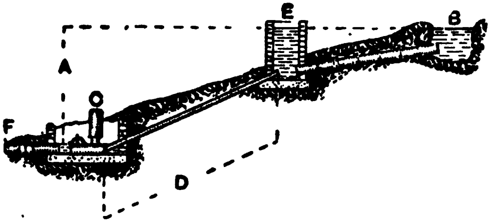


FIG. 218.
METHOD OF SETTING HYDRAULIC RAM.

Fig. 218 shows the method of fixing the hydraulic ram, A being the vertical difference in feet between the water surface at the source of supply B and the ram C; D is the drive pipe, which is of the same length as the vertical height to which the water is to be raised. As will be seen, the fall is an even gradient from the source of supply to the ram, and the distance between the ram and the source is greater than the length of the drive pipe; it will therefore be necessary to build the small brickwork tank E, the top of which must be a little higher than the surface of the water at the source. This tank is used to prevent the bursting of the pipes when the water is suddenly checked by the closing of the pulse valve, for with long pipes the pressure at the moment when the valve closes would be greater than the pipes and joints could withstand. The ram is bolted to a solid concrete foundation, and should be placed in a small house as shown. The level of the pulse must be at such a height that it is never covered by the tail-water, which is drained into a ditch at F.

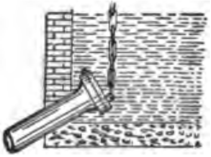


FIG. 219.
FLAP VALVE ON DRIVE PUMP.

The drive pipe should be laid at an even gradient throughout. With small rams up to 2 inches in diameter, this pipe may be of wrought iron or steel with threaded couplings, but for larger rams cast-iron flange pipes should be used of a thickness sufficient to withstand the pressure without bursting. All the joints must be water tight, and when wrought iron pipes are used, their ends should butt together in the couplings. To prevent the contraction of the column of water at its entry to the drive pipe, the end in the tank should be fitted with a trumpet mouth and flap valve, controlled by a chain, as shown in Fig. 219.

The tank and source of supply are connected by ordinary socketed drain pipe, having a diameter 2 to 3 times that of the drive pipe. The joints in the supply pipe are made with cement. A copper strainer should be fitted to the inlet end of the pipe to prevent debris from entering and choking it. The opening in this strainer should be of greater area than the pipe. As a further protection against the strainer becoming blocked by weeds and rushes collecting around it, a wire netting may be placed around it and secured to posts driven into the bottom of the stream near the pipe. The posts are driven in about 1 ft. away from the pipe on each side, and the netting fixed to allow a space of 1 ft. between the netting and the strainer.

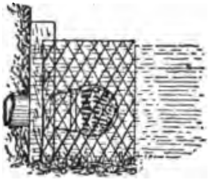


FIG. 220.
DETAIL OF STRAINER.

The ram having been fixed in position and the drive pipe and the supply pipe to the tank laid, an attempt can be made to start the ram. The starting is effected by pressing down the pulse valve so as to let the water discharge and then allow it to rise, continuing to work it in this way for a few strokes by hand, when it should continue to beat by itself if the adjustment of the valve is suitable for the conditions of the water supply and the height the water is to be raised. But more probably it will be found that the ram will not work before it is adjusted. The adjustment is effected by the two nuts and the brass washers at the end of the valve spindle.

The nuts are for adjusting the length of the beat, and the washers are for adjusting the weight of the pulse valve by adding to or taking away. The length of the beat, and weight of the valve depend upon the length of the drive pipe, and head of water. The stroke or beat of the valve should be as long as possible for efficient working.

On a slow and long stroke the ram will use and lift more water than on a fast and short one. The number of strokes is reduced and the length increased by unscrewing the nuts, and the number of strokes is increased and the length decreased by screwing them up. The weight of the pulse valve should be about the same as that of the water in the drive pipe when it is at rest. To adjust the valve to obtain the best results will occupy considerable time, as many trials will have to be made, by reducing or increasing, as required, the weight of the valve, and the number and length of the stroke, until a satisfactory result is obtained.

Stoppage of the ram after it has been working for a considerable time may be due to any of the following causes: The area of the drive pipe may be reduced or roughened by rust, or a deposit of organic matter may have formed on the inner surface of the pipe. In this case, although the head may be sufficient to dash the valve on to its seating, the force of the recoil may not be sufficient to allow the valve to open ready for another stroke. To remedy this, a bundle of wire should be dragged through the drive pipe several times, by means of a length of rope, after removing the ram from the pipe. Leaky joints in the drive pipe may also prevent the water recoiling sufficiently to open the pulse valve. The variation in level of the head water is a very common occurrence where the supply is intermittent. The level of supply being lowered, the air is carried in with the water, causing the valve to chatter for a short time, and ultimately to remain closed, though if the valve is a heavy one it will remain open. The air vessel may be water-logged, owing to the snifting valve becoming inoperative, through being covered by the tail-water; or the snifting valve may be choked, causing the resistance in the air vessel to be so great that the delivery valve will not open, thus stopping the delivery of water, though the pulse valve will continue to beat, and everything will appear to be in perfect working order. The recoil of the drive water may be insufficient to allow the pulse valve to open. With a slow lift, the absence of air in the air vessel may sometimes be detected by the water being delivered in a succession of spurts accompanied by concussion in the pipes. To remedy this, shut the flap valve on the end of the drive pipe in the tank, remove and empty the air vessel, and see that the snifting valve is in good order. On replacing the air vessel, the ram should again deliver water properly.

To stop the ram, the pulse valve is held up for a few seconds against its seating, when it will cease to beat; but to stop the ram for examination and repairs, the head or flap valve in the tank must be closed.

Efficiency.—In tests, the highest efficiency obtained from a ram was 74.9 per cent, but this varies according to the ratio of lift and fall, being greater at low than at high ratios. Clark gives the following percentages of efficiency for ratios varying from 4 to 26:

Ratio of lift to fall......	4	6	8	10	12	14	16	18	20	22	24	26
Efficiency per cent......	72	61	52	44	37	31	25	19	14	9	4	0

Thus with 10 feet head and 40 feet lift, the ram would show an efficiency of 72 per cent. But with 10 feet head and 260 feet lift, the ram could just raise water to that height, but it would be unable to deliver any; hence its work at that ratio would have no efficiency.

CONSTRUCTION OF DAMS.

In the construction of a dam across a stream of water, especially if the bottom is of an alluvial nature, the main problem is to successfully guard against the water finding its way underneath the dam instead of over it. Of course reinforced concrete construction when properly put in is the best method of solving the problem of a foundation for a dam, but it is not always possible to use this material, owing to objections on the part of the owners regarding its high cost. But when it can be used, the undermining of the structure may be prevented by a properly constructed concrete blanket extending for some distance above and below the dam. In case concrete is not used, the next best method of getting a foundation is by means of heavy timbers termed mud-sills which are imbedded in the bottom as deeply as possible.

In important cases asphalt can be used. It can be cut to solid and non-porous bottom. In any case it should be left in such a manner that the pressure of water on it after completion of the dam will not be sufficient to permit percolation into, through or under it so as to weaken or ruin the structure as a whole. Each 32 ft. in depth of water exerts one net ton pressure on each sq. ft.; water under such pressure will find a way through voids, crevices, etc., quite invisible to the human eye and apparently tight.

When possible for reasonable cost perishable or porous earth MUST be removed to a line at least 10 ft. below the down-stream side of center puddle or core wall and to a firm and water tight foundation into which should be cut parallel trenches under the entire length of the embankment; such trenches ought not to be a greater distance apart than 0.5 times the height of dam or embankment at center; they should be 4 or 5 ft. wide and about the same depth if in a good or fair foundation.

Always have trench cut just inside (under embankment) of inside foot of slope.

If foundation is "questionable" increase the depth accordingly.

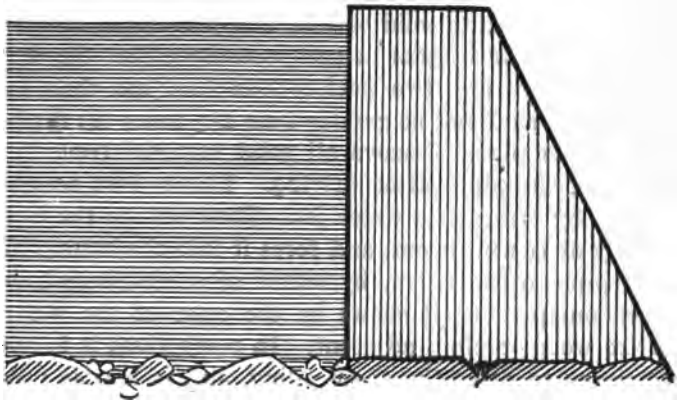


FIG. 221.

VERTICAL SECTION CONCRETE DAM.

The removed perishable earth, etc., can be placed in down-stream toe of slope, or saved for a top dressing for slope. Frost makes any earth porous; therefore remove everything to point below that which has been effected by frost.

Foundations in Porous Soil, Quick Sand, Etc.—It frequently happens that a dam must be placed in locations where a porous sand, gravel or quick-sand layer or bed must be contended with; to excavate to "solid bottom" would cost in many cases more money than was available for the project; in others it is often practically impossible to do so.

If tests show that the subsoil water is flowing in a direction parallel with the stream on the surface, drive across the valley a few feet below the center line of the dam a row of tight sheet piling; then parallel with this drive rows of two, three, or four inch perforated pipes placed a distance apart each way not greater in feet than the pipes are inches in diameter. The pipes should be driven to solid bottom, and perforated around and along their lengths in such a manner that the distance between any two perforations (not less than 3/8 inch holes) shall not be greater than 1½ times the diameter of the pipe. Remove all solid material from the inside of the pipes after driving. This is best accomplished by means of a water jet. Then mix up the best quality of cement grout, and force it into the bottom of each pipe in succession, by means of a centrifugal, or other pump, until it shows by the color of the water in the wells or pipes adjacent. In many cases it is not necessary to drive the sheet piling, still is a wise precaution in every case. Proper work will insure a foundation after the cement has had time to set, which will be sufficiently strong for the core wall of the dam. The width of the strip covered by the "wells" should be not less than twice the width of the core wall, or if a core wall is not used, the width should be not less than 10 feet.

Slope of Dam.—The inside slope of a dam ought not to be less than 1.5 to 1, even when the best of embankment material is used in construction, and it in turn properly paved or covered with rip-rap. Unless it is protected by rip-rap or other paving, wave action will in time reduce the slope to from 4 or even 10 to 1, depending on the material used.

The greater the slope, the more permanent will be the dam.

THE ELECTRIC MOTOR IN MILLWRIGHTING.

Considered from a mechanical standpoint only, there is little choice in point of reliability between a direct-current motor, and an alternating-current induction motor. The direct-current motor armature has the mechanically undesirable commutator and brushes and, usually, a larger proportion of insulating material than the alternating-current motor which, however, labors under the inherent mechanical disadvantages of a very small air-gap or clearance for the rotating element.

The particular method of application selected for any type of motor has more direct bearing on the durability and life of the motor than is commonly supposed. This can be appreciated when it is remembered that the revolving armature of any type of motor is composed of a great number of parts, coils of wire, etc., held together by bands or wedges, and depending to a greater or less extent on fibrous or cotton insulating materials of little mechanical strength and, therefore, more susceptible to injury from jars and vibration than other forms of revolving machinery. It is important, therefore, when considering a motor application, to choose a method of connection which will tend to absorb or nullify these effects unless the motor has been designed with a special aim to meeting these conditions, as in railway motors, special mill motors, etc.

Belt Connection.—Wherever conditions will allow, belt drive should be used, as the belt forms the most flexible connection between motor and load and is the best-known absorbent of mechanical shocks due to load variations. For ideal conditions the belt speed should be from 4,000 to 5,000 feet per minute.

In general the load per inch width for single, or light double belts should not greatly exceed 40 pounds for belt speeds under 4,000 feet per minute and should not exceed 32 pounds for speeds from 4,000 to 6,000 feet, which is the practical limit of working. The total working pull on belt may be found from the following equation:

$$\text{Pull in pounds} = \frac{\text{hp} \times 33{,}000}{\text{Belt speed in feet per min.}}$$

Belt speed in feet per min. = Pulley dia. in inches × 0.262 × r.p.m.

For certain classes of service where belted motors operate in dusty locations, as in clay crushing plants, cement mills, flour mills, etc., the action of dust accumulations on the belt tends to reduce its pulling power and necessitate an increase in width of from 25 to 50 per cent over that which would be satisfactory for ordinary conditions. Experience has shown that a four-ply treated canvas or rubber belt is approximately equal in transmitting power to a single or light double leather belt of good quality.

The maximum ratio between driving and driven pulleys should not exceed 6:1 for ordinary conditions, and the distance between centers will depend on the ratio, good proportions being approximately as follows:

Approx. Ratio.	Min. Distance Between Centers
2:1	8 feet
3:1	10 feet
4:1	12 feet
5:1	15 feet
6:1	20 feet

All belts should have a normal lap on the pulleys of at least 160 degrees unless some form of idler pulley is used to increase belt contact. The use of paper pulleys allows a somewhat smaller degree of lap owing to the better adhesive qualities of this material, and for this reason is to be recommended for all small pulley drives.

It is not usually considered good practice to use heavy double belts of any kind on pulleys smaller than 15 inches diameter, as the abrupt bending of the thick belt over a small pulley soon opens the fibers and ruins the material.

Rope Connection.—On a par with belt driving in its effect on the motor can be placed rope drives of either the English or American system, and many installations are at present using this drive for large motors. The English or separate rope system is a favorite in rolling mill or other work where large powers are transmitted and continuity of operation is especially desirable, while the American or single rope system is especially adapted to group drive from a large unit. For either system as large a rope as possible should be chosen, for economical reasons.

To insure reasonable life of the ropes, the smallest sheave over which the ropes run should be not less than forty-two times the diameter of the rope used. Lubricated core rope should be used, as the principal friction tending to destroy rope is between the fibers themselves and, owing to the action of centrifugal force, external lubricant will not penetrate the rope.

Considering the life of the rope alone, experience shows that the most economical speed is 4,500 feet per minute, and this should be used where cost of maintenance is an important factor.

Very small pulleys or sheaves, in general, and on induction motors in particular, should be avoided, as they necessitate heavy belt tension to transmit the power and invariably produce bearing troubles which militate greatly against efficient operation, and on induction motors with their air-gap clearance cause secondary troubles liable to damage the machine seriously.

Toothed Chain Connection.—In many applications where positive connection is desirable, a toothed chain, such as the Morse or Renolds type, offers a somewhat flexible and a very satisfactory method of motor connection. When the center distances between driving and driven shafts are too short for belts and too great for the use of gears, the maximum ratio obtainable with chain is approximately 6:1. As short a center distance should be used as possible owing to the expense of the chain. For the best operation, the chain speed should not exceed 1,500 feet per minute, as beyond this speed the necessary grease for lubrication is thrown off and the wear becomes excessive. It is also imperative that means be provided for adjusting the sprocket center distances to take up the wear in chain and sprockets, and keep same fairly tight. Chains work best when driving horizontally. The driving side should be on the bottom when distance is long, and on top when distance is short or speed high. For certain very severe conditions spring center gear sprockets have been used, and found very satisfactory in protecting the motor and increasing the life of the chain. Chain drive imposes a more severe strain upon the revolving parts of a motor than a belt drive does, and this fact must be borne in mind and should be properly discounted in the type and design of motor recommended for a specific application.

Gear and Pinion Connection.—Owing to its economic advantages as to floor space, etc., the trend of new development appears to be in the direction of direct geared applications, and it is where this condition exists that the greatest precautions must be taken to eliminate those undesirable factors which usually result in a rapid depreciation of the motor, unless it has been designed with a view to fitting this particular application.

In general it has been found that where standard motors designed for belted service, have been used in general applications, driving roll trains, bending rolls, plunger pumps, glass grinding machinery, cement mill machinery, clay working machinery, coal and ore handling machinery, hoists, cranes, etc., etc., failure has been very frequent where the mechanical limitations of the electric motor were not properly understood and taken into account.

As an engineering proposition, the use of a standard commercial motor of over 20 horse-power with a pinion mounted directly on the motor shaft should be avoided wherever possible. Where service conditions demand a direct-geared connection, an ideal arrangement, and one to be used wherever possible, is to have the pinion carried on a short shaft in separate pedestals or other bearings; this shaft to be coupled to the motor shaft by some form of flexible or cushion coupling. This arrangement at once removes the motor from the destructive vibrations and hammering of the gears, allowing a lighter type of motor to be used, and, while somewhat more expensive than the common method of mounting the pinion directly on the motor shaft, the increase in cost is a cheap premium to pay for the practically absolute continuity of operation assured. It is especially applicable in many lines of industrial work where wear on the bearings of the driven mechanism is an unimportant item, and where the motor bearings are liable to be neglected.

Where the above arrangement is not permissible and where operating conditions are not too severe, an outboard bearing for the motor shaft beyond the pinion is to be recommended for motors of 20 horse-power and over, as this auxiliary increases by nearly 100 per cent the resistance of the shaft to bending, and the ability of the bearings to resist pounding out. As the chief cause of trouble on geared motors can be traced directly to the hammering and vibration due to the tooth impacts, the best practice demands that for commercial motors, where the pin is mounted directly on the shaft, speeds should not exceed 700 R.P.M. for motors above 20 horse-power, and for heavy work even this is too high.

The maximum pitch line speed at which cut metal gears should be run is approximately 2,000 feet per minute, and speeds higher than 1,200 feet are not to be recommended for best results, particularly where the pinion is mounted directly on the motor shaft.

Pitch line speed may be determined from the following equation:

$$\text{P.L.S.} = \text{Rev.} \times \text{Pitch. Dia. in inches} \times 0.262.$$

$$\text{Pitch Dia. in inches} = \frac{\text{Number of teeth}}{\text{Diametral Pitch}}$$

$$\text{Pitch Dia. in inches} = \frac{\text{Number teeth} \times \text{Circular Pitch}}{3.1416}$$

As in ordinary motor drives the pinions are usually of forged steel, the strength of the gear only need be considered when figuring the size of gear required. Table 37 shows a ready means of determining the proper proportions of cast iron gears having more than 25 teeth to transmit a given horse-power, also the gear pitches preferable for motor drives.

TABLE 37.—SHOWING VALUES OF HP PER INCH WIDTH FOR PITCH LINE SPEEDS ABOVE 1,000 FEET PER MINUTE.*

HP. to be Transmitted	Preferable Diametral Pitch	HP. per Inch Width per 100 ft. per Min.
1 to 2	8	0.24
2 to 5	6	0.30
5 to 10	5	0.38
10 to 30	4	0.48
30 to 75	3	0.64
75 to 125	2½	0.76
125 to 200	1¾	1.10

* For cast steel gears the values in HP per inch width should be increased 2½ times.

For pitch line speeds from 500 to 1,000 feet per minute the above values of horse-power should be increased 50 per cent, and for speeds up to 500 feet per minute should be increased 100 per cent.

The minimum pitch diameter of motor pinion is approximately twice the diameter of shaft for ordinary cases, and the smallest pinion should not have less than 14 teeth under any condition.

Example:—25 horse-power motor, 600 R.P.M., countershaft speed 100 R.P.M.

Diameter motor shaft at pinion fit = 2½ inches.

Smallest pitch diameter possible = 2 × 2½ = 5 inches.

From table of preferable pitches, pitch = 4 inches.

Therefore, smallest number of teeth = 4 × 5 = 20.

Pitch line speed = 600 × 5 × 0.262 = 786 feet per minute.

$$\text{Horse-power per 100 feet per min.} = \frac{25 \times 100}{786} = 3.18.$$

From the table it is evident that horse-power per inch of width at this speed = 0.48 + (50 per cent) = 0.72.

$$\text{Therefore, the proper face of cast iron gear} = \frac{3.18}{0.72}$$

= 4.41, or, say, 4½ inches.

$$\text{Proper number of teeth for gear} = \frac{600 \times 20}{100} = 120$$

teeth.

Where cast iron gear faces figure out greater than seven inches, it will be economical to use cast steel gears, as by their use the face may be made one-half the size.

When high speed motors must be used in geared-applications, experience has shown that rawhide pinions alleviate vibrations, noise, etc., to some extent, although not sufficiently to make unnecessary such precautions as should be taken where steel pinions are used. The value of the rawhide lies principally in its noiselessness of operation, especially where the rawhide is not shrouded, viz., where the design is such that the retaining brass checks are cut below the roots of the teeth. With such forms of pinion, pitch line speeds of 3,000 feet per minute have been used successfully on motors up to 50 horsepower. The strength of rawhide is approximately the same as cast iron, and this fact must be borne in mind when considering the use of rawhide pinions, as it will be evident that a steel gear cannot be driven to its full capacity by a rawhide pinion.

For a majority of motor applications, involute forms of cut gear teeth are to be recommended, as this form does not require the maintenance of exact center distances demanded by other forms of teeth. Also, because this form has been adopted as standard by the majority of gear manufacturers under the diametral pitch system.

Cushion Couplings.—In many applications, such as air compressors, plunger pumps, etc., the motor is required to be coupled directly to an intermediate shaft which usually carries a pinion for transmitting power to other parts of the mechanism. Application of a motor to this service should be through a cushion coupling for best results, as with the usual forms of rigid coupling the shocks and vibrations from the gears due to the cyclical variations in the load are transmitted back to the motor and are very destructive to the revolving windings. Vertical motors driving centrifugal pumps through long vertical shafts should always be connected together by means of a flexible coupling that will allow considerable displacement in shaft alignment as, owing to the absence of gravitational forces in the plane of revolution, all vertical running motors and shafts are extremely sensitive and destructive vibrations are easily started.

Flange Couplings.—For installations where the motors have to deliver a practically constant torque, and where there is little liability of shafts getting out of alignment, such as centrifugal pumps, blowers, motor-generator sets, etc., a flange or rigid form of coupling has been found entirely satisfactory. A careful investigation should always be made, however, of all factors liable to influence operation, as successful application of this form of coupling requires practically ideal conditions.

Foundations.—In many motor installations insufficient attention and care are bestowed on the motor foundation, or method of mounting to insure satisfactory motor operation. This is particularly true of geared applications where motors are frequently found shimmed up with wooden blocks. This should not be tolerated, as it can be nothing else than unsatisfactory; rigidity of mounting or foundation being of extreme importance in geared applications.

PIPES AND PIPE FITTING.

As has been previously stated, it frequently happens that the millwright is called upon to superintend the laying out and erection of the steam and water piping systems for a mill or factory, and considering this to be the case, a space will be devoted to the discussion of this most important subject.

Selection of Pipe.—This is an important matter, and care and good judgment should be exercised in the choice of pipe that is to be used on the job. The bulk of the pipe manufactured today is steel, it having been demonstrated by experiment and also by practice that steel pipe is far superior to iron pipe for durability, and hard usage, consequently it has come to stay. Therefore one of the requirements of the successful pipe fitter is to be able to distinguish steel pipe from iron pipe, and a few pointers on this subject will be presented.

In the first place, iron pipe is rough in appearance and the scale on it is heavy, whereas on steel the scale is very light, and has the appearance of small blisters or bubbles, underneath which the surface is smooth and somewhat white. When flattened, steel pipe seldom breaks; but if a fracture does occur, it will be noticed that the grain is very fine. Iron pipe, when subjected to this test, breaks readily and shows a coarse fracture, due to the long fiber of this material.

The impression often prevails that steel pipe is exceedingly hard, for which reason they imagine that it is threaded with difficulty and that the threads are easily broken off. This belief is entirely erroneous, the truth being that steel pipe is soft and tough. Threads on this pipe do not break; they tear off, to avoid which it is necessary that the cutting die shall be sharp and thus cut above the center. Dies suitable for steel pipe can also be used on iron pipe; but blunt dies that will work successfully on iron pipe will tear the threads on steel pipe, owing to the softness of the metal.

Laying Out a System of Piping.—A good rule to follow in laying out a piping system is as follows: If for steam, start at the distributing end of the system, and work toward the boilers, gradually increasing the size of the pipe at each outlet until all have been provided for, and the total area of which should not exceed the area of the steam main leading from the boilers. For water the same rule will apply, always bearing in mind that, at the same velocity of flow the capacities of pipes are proportional to the squares of their internal diameters, as for instance one 4-inch pipe will deliver the same volume as four 2-inch pipes. With the same head or pressure, however, the velocity is less in the smaller pipe, and the volume delivered varies about as the square root of the fifth power (*i. e.*, as the 2.5 power).

Table 38 will be found very convenient in calculating sizes of pipes, and the number of smaller sized pipes required to equal in capacity one larger.

The method of using the table is as follows: Assume the main or header leading through the factory to be a 4-inch pipe. Reference to Table 38 shows that the internal area of a 4-inch pipe equals 12.730 square inches. From this header the following branches may be safely led:

One 2-in. pipe, internal area = 3.356 sq. in.
Two 1½-in. pipes, internal area = 4.072 sq. in.
Two 1¼-in. pipes, internal area = 2.992 sq. in.
Two 1-in. pipes, internal area = 1.722 sq. in.

Total combined area = 12.142 sq. in.

and 12.730 — 12.142 = .588 sq. in. as a margin.

For the smaller sizes of branch pipes, say from 1 in. down to ½ in., which are likely to be tapped into the above mentioned pipes, the table may again be consulted, as for instance, from the 2-in. pipe having an area of 3.356 sq. in., there may be taken:

One 1¼-in. pipe, internal area = 1.496 sq. in.
Two ¾-in. pipes, internal area = 1.066 sq. in.
Two ½-in. pipes, internal area = .608 sq. in.

Total combined area = 3.170 sq. in.

and 3.356 — 3.170 = .186 sq. in. as a margin.

Table 38 follows:

TABLE 38.—INTERNAL AREA OF WROUGHT IRON OR STEEL WELDED PIPE.

Nominal Inside Diameter Inches	Internal Area Sq. Inches	Nominal Inside Diameter Inches	Internal Area Sq. Inches
⅜	.191	5	19.986
½	.304	6	28.890
¾	.533	7	38.738
1	.861	8	50.027
1¼	1.496	9	62.730
1½	2.036	10	78.823
2	3.356	11	95.033
2½	4.780	12	113.098
3	7.383	13	137.887
3½	9.887	14	159.485
4	12.730	15	182.665
4½	15.961		

Ordinary or standard black pipe, from ⅛ to 1 inch inclusive, is butt welded, and tested to 300 pounds to the square inch. Pipe 1¼ inches and larger is lap welded, and proved to 500 pounds to the inch, and the millwright should keep well within these figures when handling steam or water under heavy pressures. There is also made and sold from stock a heavier pipe known as "hydraulic." This pipe should be used when very heavy pressures are to be resisted, particularly when there are shocks similar to those when the hydraulic ram is used.

Expansion and Contraction.—In laying out a system of steam-piping for a power plant, perfect freedom for expansion and contraction should be allowed, to prevent undue strains on any member of the system, or at the joints, causing them to leak. The old types of slip-expansion joints having proved a constant source of trouble and expense requiring frequent repacking and adjusting, are seldom, if ever, used on a good job of piping. If absolutely necessary, however, to use this type of expansion joint, the piping should be so anchored as to prevent the joint from pulling apart.

With the advent of higher steam pressures and correspondingly higher temperatures and velocities, more attention has been given to the proper designing of piping systems, and constructive details than in the past. Steel pipe bends of long radius are used wherever practical in place of the cast elbows of short radius. They take up the expansion stresses, making the system more flexible throughout; reduce vibrations and friction, and deliver the steam to the engine with a lower drop in pressure.

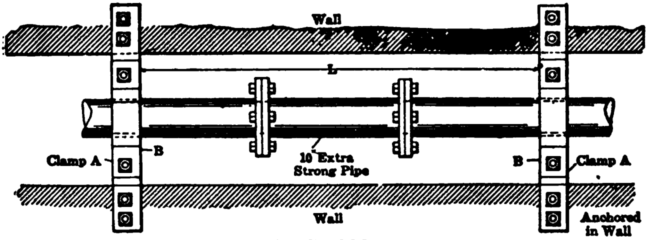


FIG. 222.

Pipe bends curved to a radius of less than five diameters of the pipe are undesirable as expansion bends. The radius should be at least five or six pipe diameters.

Anchoring.—This is an important detail, especially in large sized pipe, and in order to be able to correctly locate the anchors, the designer should possess a good knowledge of the expansion of pipes when heated, and their contraction when cold. Fig. 222 shows a 10-inch extra heavy steam pipe, and two clamps, A A, anchored rigidly to the wall. Assume the pipe to have been placed in position while at a temperature of 60° Fahr. and held securely in one clamp, and allowed to move freely through the other one while expanding. Now if while in this condition, steam at 250 lbs. pressure, and superheated 100 degrees is turned into the pipe, the elongation of the pipe due to expansion will be $1\frac{7}{8}$ in. in each 60 feet of length; that is, the pipe which was originally 60 feet long, is now 60 feet $1\frac{7}{8}$ inches in length, due to the added temperature of 441 degrees, and the movement of the pipe through the loose clamp is $1\frac{7}{8}$ inches. But suppose the pipe to have been rigidly clamped between both clamps, with no chance for endwise movement during the process of heating, the result would be that either the clamps would spring, or the pipe would buckle, or bend sideways, owing to its great length as compared with its diameter. The tremendous force exerted upon the material, and joints of the pipe by the thrust of expansion, and the pull of contraction may be calculated by means of the following formula: $P = C\,T\,A\,E$ in which $P =$ thrust in lbs. from expansion, or pull due to contraction. $C =$ co-efficient of linear expansion $= 0.000006$. $T =$ difference in degrees Fahr. of temperature between original temperature and temperature after pipe has been heated, or cooled $= 441$ degrees. $A =$ area of metal in cross section of pipe in square inches which for 10-inch extra heavy pipe $= 16$. $E =$ co-efficient of elasticity $= 30{,}000{,}000$. By substituting these values we have: $P = 0.000006 \times 441 \times 16 \times 30{,}000{,}000 = 1{,}270{,}080$ pounds.

These figures give some idea of the strains to which fittings and joints are subject where expansion and contraction are not properly provided for in steam lines, or where anchors are improperly located. As another illustration of the strains exerted upon the fiber of the pipe, and upon joints and fittings, suppose that the pipe already referred after being heated and allowed to expand freely, is then rigidly clamped at both ends, and allowed to cool off again to its original temperature, the tensile strain upon the material in the pipe would be:

$$\frac{P}{A} = \frac{1{,}270{,}080}{16} = 80.000 \text{ lbs. per square inch,}$$

which would cause rupture.

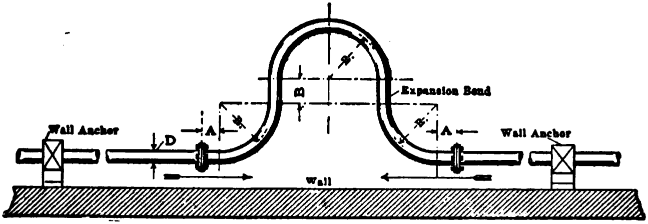


FIG. 223.
FOR LONG RUNS.

Allowing for Expansion and Contraction.—It is considered good practice in figuring for expansion to allow only one-half the calculated amount when cutting the pipe to length. For example, if in a run of pipe 100 feet between connections, or points where steam lines are taken off from the header, the expansion is calculated to be 3 inches; allow 1½ inches when cutting the pipe to length, or in other words, the total length of pipe should be 100 — 1½ = 99 feet 10½ inches.

Then the steam fitter takes up the other 1½ inches when erecting the line, with the result that when steam is turned on the expansion removes the tension or strain put on the pipe when cold, and the fittings and joints are strained only one-half as much as if none, or all of the expansion were allowed for. This rule is used by most large concerns, and has proven to be satisfactory to operating conditions in general.

The expansion bend shown in Fig. 223 is intended for use in a long run of piping where it is desirable to anchor the line at two points to take the strain off the fittings and joints outside the anchors, not shown. When heated, the pipe expands in the direction of the arrows,

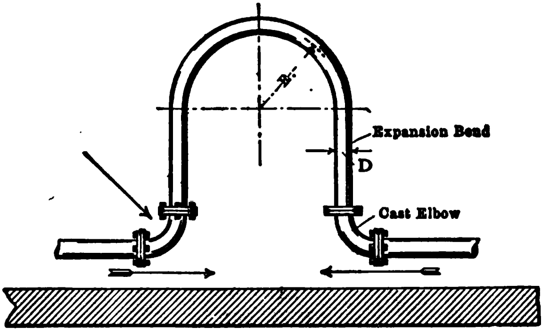


FIG. 224.

tending to close the bend. To insure sufficient elasticity the dimension B should be made as large as possible. The dimensions A and R can be respectively 5 and 1 of D.

For sizes above 6 inches it may be impossible to make this bend in one piece, owing to the long length of pipe required. Commercial pipe averages from 16 to 20 feet in length, although lengths up to 24 feet are kept in stock by some dealers for use in making up special bends.

For bends 8 inches in diameter and larger, the arrangement shown in Fig. 224 may be used for the same purpose. With such an arrangement, however, a bending strain is put on the joints, as shown in Fig. 225, which is greatly enlarged to show the straining action on the joint at E, as the pipe expands.

The cast elbows should be extra heavy, with thick flanges. The flanges on the bend should be thick also, as a thin flange is easily ruptured or strained sufficient to cause leakage at the joints.

Where there is sufficient room, a better arrangement would be to substitute two 90-degree or square bends of steel pipe in place of the cast elbows, making the system more flexible.

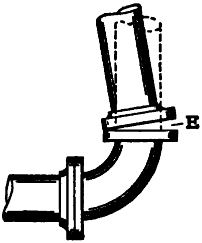


FIG. 225.

One method of anchoring a steam header of moderate length is shown in Fig. 226. The line of pipe which may be from 50 to 150 feet in length is securely anchored at the middle of its length, and allowed to expand in both directions, as shown by the arrows A A. The joints nearest the anchor or clamp suffer the least strain, while those joints nearer the ends of the header are subject to the greatest strains from expansion and contraction In laying out a system of piping, the amount of expansion for the different runs should be worked out, and provision made accordingly. The expansion of cast iron in fittings is about the same as untempered steel, but under the strain and stress of superheated steam (combined heat and pressure), a fitting 22 inches in length from face to face of flanges has been known to have a permanent elongation of $\frac{5}{16}$ in. in four years' use. Therefore, in the design of steam pipe lines, the question of superheat must be considered as tending toward the permanent elongation, and consequent loss of tensile strength of the material.

Fittings and Valves.—For flanged fittings there are two recognized standards: one for pressures up to 125 pounds, adopted by the A. S. M. E., and another for pressures up to 250 pounds, adopted by the manufacturers. Some manufacturers list their pipe and fittings as "standard" for pressures up to 125 lbs., "full weight" for pressures from 125 lbs. to 175 lbs., and "extra heavy" for pressures higher than 175 lbs.

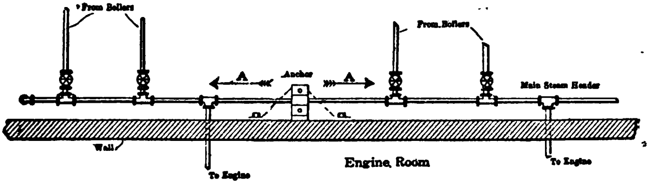


FIG. 226.

Flanged valves are made with the thickness of flange to correspond with thickness of pipe flange, and the "standard," "medium," or "extra heavy" correspond with the same grades of pipe and pipe flanges.

It is a generally accepted fact that few, if any, of the accidents caused by the failure of pipe fittings are due only from the pressure of the gas or fluid contained in them, but are the result of combined stresses caused by the expansion and contraction of the pipe, water hammer, or pipe improperly supported, any or all of these being in conjunction with a stress due to the pressure of the gas or fluid within the fitting.

A fitting may, therefore, be subjected to two classes of stresses: one, a legitimate stress, due to the pressure from that which the fitting contains, such as water, steam, compressed air, etc., and the other, due to strains caused by systems of piping improperly designed or installed. Stresses due to the former are tangible and may be estimated with some degree of accuracy, but stresses due to the latter are an unknown quantity varying with the wisdom of the designer and for which the factor of safety must provide.

The dimensions of fittings of different makers vary slightly, so that what is true of certain sizes of one maker's fittings might not be true of another's.

Table 39 gives the bursting strength of standard cast iron fittings, screwed. It will be seen that the elbows show a greater strength for all sizes than do the tees. Table 40 gives the ultimate test pressure of malleable iron pipe fittings including both black, and galvanized. The bodies of these fittings were .25 in. in thickness, and the average bursting pressure of the black fittings was 2,900 lbs., while that of the galvanized fittings was 2,833 lbs. These important details concerning pipe fittings were obtained through a series of tests conducted by Mr. S. M. Chandler at the laboratories of a prominent valve and fittings manufacturing company.

TABLE 39.—STANDARD SCREWED CAST-IRON ELBOWS AND TEES—ULTIMATE TEST PRESSURES.

Size	Elbows		Average	
2¼	3500	3300	3400	3400
3	2400	2600	2100	2500
3½	2100	1700	2400	2250
4	2800	2500	2500	2600
4½	2000	2600	2600	2600
5	2600	2500	2500	2533
6	2600	2200	2300	2367
7	1800	2100	1900	1950
8	1700	1600	1700	1667
9	1800	1800	1900	1833
10	1800	1700	1600	1700
12	1100	1200	900	1150

Size	Tees		Average	
1¼	3400	3300	3300	3333
1½	3400	3200	2800	3300
2	2500	2800	2500	2600
2½	2400	2100	2500	2450
3	1400	1900	1800	1850
3½	1200	1500	1800	1650
4	1800	2100	1700	1867
4½	1100	1400	1400	1400
5	1700	1300	1500	1600
6	1400	1500	1100	1450
7	1400	1400	1500	1433
8	1200	1400	1300	1350
9	1300	1400	1200	1300
10	1100	1300	1200	1200
12	1100	1000	1100	1067

TABLE 40.—BURSTING PRESSURE OF MALLEABLE PIPE FITTINGS.

Test No.	Black or Galvanized	Ultimate Test Pressure	Weight, Pounds
1	Black	2800 lb.	10.06
2	Black	3100 lb.	10.12
3	Black	2800 lb.	10.19
4	Galvanized	2700 lb.	10.62
5	Galvanized	3000 lb.	10.50
6	Galvanized	2800 lb.	10.75

Testing for Leaks.—When the system is completed, a test should be made for leaks, particularly obscure leaks. Leaks in water piping will quickly manifest themselves when the pressure is turned on, but for steam pipes some little time will be required for the steam to condense in the pipes, when the first intimation of a leak will be water dripping from it. Very often there are steam leaks which, owing to the dryness of the steam, may be heard but not easily located. A good way to search for leaks is to prepare a pail full of strong soap-suds, and using a paint brush go over the lines of pipe, applying the mixture to each one of the joints. If there is even a very slight leak any place where the soap-suds touch, bubbles will be blown, and the leak located. It should then be marked with a piece of white chalk carried along for the purpose.

Air and Water Traps.—When running pipe lines for steam, the great thing to be avoided is the forming of pockets in which water may collect and thus cut down the capacity of the pipe. Under certain conditions, the pipe may become completely filled with water in spots and the flow of steam cut off until a lowering of pressure at the discharge end is sufficient to overcome the hydrostatic pressure of the collected water and that substance is driven out of the pipe. Water traps or "pockets" in a steam pipe are always to be avoided. They are very undesirable, and in some instances they become dangerous.

Water pipes are also cut down in capacity by pockets which may contain air. But in this case, instead of a low place in the pipe giving trouble, the high spots are to be avoided. A collection of air in the bend of a pipe passing over a high place in the line may, in many instances, cut down the pressure greatly, and in some cases entirely stop the flow of water. Particularly is this the case when the pipe acts as a syphon; the pressure or weight of the water in the discharge leg must overcome the weight of water in the supply leg of the syphon.

In water pipes under pressure, the action is the same, though the force not being limited to 14.7 pounds to the square inch can overcome to better advantage the presence of air in the pipe. In either case, the loss of pressure amounts to the loss of head equal to the height of column of air in the pipe, calculated for an equal height of water. Thus, pockets and traps must be avoided in both steam and water piping, and the straighter the pipes can be run, in a vertical direction, the better will the pipe line serve its purpose.

Dead-Ends and Drips.—Dead-ends in pipe lines should always be avoided if possible. No matter where placed, a dead-end always fills with water. If the pipe be a large one and the dead-end projects vertically upward, the water may run out as fast as it collects, but unless a dead-end is drained in some way, it will surely fill with water—and stay filled. The manner of its filling is through the condensation of the steam which flows to the dead-end. A pipe with a close end is much like an extended bellows. The condensing of the steam in the pipe is like the closing of the bellows. As everything—boards and leather—is drawn down upon the bottom board of the bellows, so every bit of condensed water is drawn to the dead-end of a steam pipe by the condensing of the steam in that portion of the pipe.

When it is required to maintain a circulation in a dead-end, a drip should be provided which will take care of the water of condensation as fast as it collects. Either a drip must be established or a loop must be made, whereby what would be the dead-end is connected to some portion of the steam system by means of which the water of condensation may pass away. Hand operated drips may be used, but this is at the best, a wasteful and unreliable method of disposing of the condensation.

A much better method is the employment of automatic traps, discharging into a return main which will carry the water of condensation back to a receiver in the boiler room or pump room, from whence it may be taken by the feed pump and returned to the boilers.

Making Up Pipe Joints.—For making up permanent screw joints in pipe, where it is not necessary to break the joints again, nothing is better than good red lead, though litharge (yellow oxide of lead) answers nearly as well. For screw joints which may have to be taken down frequently, plumbago mixed with cylinder oil, makes a good material for daubing either threads or flanges; it is also good on flange bolt threads. For threaded pipe, use as thick as possible. A mixture which will run is too thin. It daubs things too much. For flange use, it may be mixed thin enough to allow of being spread on the surfaces with a brush. When nice work is required, daub the inside of fittings, valves, etc., and it will not show as when daubed on pipe, nipples, etc.

USEFUL INFORMATION.

One heaped bushel of anthracite coal weighs from 75 to 80 lbs.

One heaped bushel of bituminous coal weighs from 70 to 75 lbs.

One bushel of coke weighs 32 lbs.

Water, gas and steam pipes are measured on the inside.

One cubic inch of water evaporated at atmospheric pressure makes 1 cubic foot of steam.

A heat unit known as a British Thermal Unit raises the temperature of 1 pound of water 1° Fahr.

For low pressure heating purposes, from 3 to 8 pounds of coal per hour is considered economical consumption, for each square foot of grate surface in a boiler, dependent upon conditions.

A horsepower is estimated equal to 75 to 100 square feet of direct radiation. A horsepower is also estimated as 15 square feet of heating surface in a standard tubular boiler.

Water boils in a vacuum at 98° Fahr.

A cubic foot of water weighs 62½ pounds, it contains 1,728 cubic inches or 7½ gallons. Water expands in boiling about one-twentieth of its bulk.

In turning into steam water expands 1,700 times its bulk, approximately 1 cubic foot of water will produce 1 cubic foot of steam.

One pound of air contains 13.82 cubic feet.

It requires 1½ British Thermal Units to raise one cubic foot of air from zero to 70° Fahr.

At atmospheric pressure 966 heat units are required to evaporate one pound of water into steam.

A pound of anthracite coal contains 14,500 heat units.

One horsepower is equivalent to 42.75 heat units per minute.

One horsepower is required to raise 33,000 pounds one foot high in one minute.

To produce one horsepower requires the evaporation of 2.66 pounds of water.

One ton of anthracite coal contains about 40 cubic feet.

One bushel of anthracite coal weighs about 86 pounds.

Heated air and water rise because their particles are more expanded, and therefore lighter than the colder particles.

A vacuum is a portion of space from which the air has been entirely exhausted.

Evaporation is the slow passage of a liquid into the form of vapor.

Increase of temperature, increased exposure of surface, and the passage of air currents over the surface, cause increased evaporation.

Condensation is the passage of a vapor into the liquid state, and is the reverse of evaporation.

Pressure exerted upon a liquid is transmitted undiminished in all directions, and acts with the same force on all surfaces, and at right angles to those surfaces.

The pressure at each level of a liquid is proportional to its depth.

With different liquids and the same depth, pressure is proportional to the density of the liquid.

The pressure is the same at all points on any given level of a liquid.

The pressure of the upper layers of a body of liquid on the lower layers causes the latter to exert an equal reactive upward force. This force is called buoyancy.

Friction does not depend in the least on the pressure of the liquid upon the surface over which it is flowing.

Friction is proportional to the area of the surface.

At a low velocity friction increases with the velocity of the liquid.

Friction increases with the roughness of the surface.

Friction increases with the density of the liquid.

Friction is greater, comparatively, in small pipes, for a greater proportion of the water comes in contact with the sides of the pipe than in the case of the large pipe. For this reason mains on heating apparatus should be generous in size.

Air is extremely compressible, while water is almost incompressible.

Water is composed of two parts of hydrogen, and one part of oxygen.

Water will absorb gases, and to the greatest extent when the pressure of the gas upon the water is greatest, and when the temperature is the lowest, for the elastic force of gas is then less.

Air is composed of about one-fifth oxygen and four-fifths nitrogen, with a small amount of carbonic acid gas.

To reduce Centigrade temperatures to Fahrenheit, multiply the Centigrade degrees by 9, divide the result by 5, and add 32.

To reduce Fahrenheit temperature to Centigrade, subtract 32 from the Fahrenheit degrees, multiply by 5 and divide by 9.

To find the area of a required pipe, when the volume and velocity of the water are given, multiply the number of cubic feet of water by 144 and divide this amount by the velocity in feet per minute.

Water boils in an open vessel (atmospheric pressure at sea level) at 212° Fahr.

Water expands in heating from 39 to 212° Fahr., about 4 per cent.

Water expands about one-tenth its bulk by freezing solid.

Rule for finding the size of a pipe necessary to fill a number of smaller pipes.—Suppose it is desired to fill from one pipe, a 2, 2½ and 4-inch pipe. Draw a right angle, one arm 2 inches in length, the other 2½ inches in length. From the extreme ends of the two arms draw a line. The length of this line in inches will give the size of pipe necessary to fill the two smaller pipes—about 3¼ inches. From one end of this last line, draw another line at right angles to it, 4 inches in length. Now, from the end of the 2-inch line to the end of the last line draw another line. Its length will represent the size of pipe necessary to fill a 2, 2½ and 4-inch pipe. This may be continued as long as desired.

Discharge of Water.—The amount of water discharged through a given orifice during a given length of time and under different heads, is as the square roots of the corresponding heights of the water in the reservoir above the surface of the orifice.

Water is at its greatest density and occupies the least space at 39° Fahr.

Water is the best known absorbent of heat, consequently a good vehicle for conveying and transmitting heat.

A U. S. gallon of water contains 231 cubic inches and weighs 8 1/3 pounds.

A column of water 27.67 inches high has a pressure of 1 pound to the square inch at the bottom.

Doubling the diameter of a pipe increases its capacity four times.

A cubic foot of anthracite coal averages 50 pounds. A cubic foot of bituminous coal weighs 40 pounds.

Weights.

One cubic inch of water weighs....	0.036	pounds
One U. S. gallon weighs..........	8.33	"
One Imperial gallon weighs	10.00	"
One U. S. gallon equals...........	231.00	cubic inches
One Imperial gallon equals........	277.274	" "
One cubic foot of water equals.....	7.48	U. S. gallons

Liquid Measure.

4 Gills make 1 Pint
2 Pints make 1 Quart
4 Quarts make 1 Gallon
31½ Gals. make 1 Barrel

To find the area of a rectangle, multiply the length by the breadth.

To find the area of triangle, multiply the base by one-half the perpendicular height.

To find the circumference of a circle, multiply the diameter by 3.1416.

To find the area of a circle, multiply the diameter by itself, and the result by .7854.

To find the diameter of a circle of a given area, divide the area by .7854, and find the square root of the result.

To find the diameter of a circle which shall have the same area as a given square, multiply one side of the square by 1.128.

To find the number of gallons in a cylindrical tank, multiply the diameter in inches by itself, this by the height in inches, and the result by .34. To find the number of gallons in a rectangular tank, multiply together the length, breadth and height in feet, and this result by 7.4. If the dimensions are in inches, multiply the product by .004329. To find the pressure in pounds per square inch, of a column of water, multiply the height of the column in feet by .434.

To find the head which will produce a given velocity of water through a pipe of a given diameter and length: Multiply the square of the velocity, expressed in feet per second, by the length of pipe multiplied by the quotient obtained by dividing 13.9 by the diameter of the pipe in inches, and divide the result obtained by 2,500. The final amount will give the head in feet.

Example.—The horizontal length of pipe is 1,200 feet, and the diameter is 4 inches. What head must be secured to produce a flow of 3 feet per second?

$$3 \times 3 = 9;\ 13.9 \div 4 = 3.475.$$
$$9 \times 1{,}200 \times 3.475 = 37{,}530.$$
$$37{,}530 \div 2{,}500 = 15 \text{ ft.}$$

To find the velocity of water flowing through a horizontal straight pipe of given length and diameter, the head of water above the center of the pipe being known: Multiply the head in feet by 2,500, and divide the result by the length of pipe in feet multiplied by 13.9, divided by the inner diameter of the pipe in inches. The square root of the quotient gives the velocity in feet per second.

To find the head in feet, the pressure being known, multiply the pressure per square inch by 2.31.

To find the contents of a barrel.—To twice the square of the largest diameter, add the square of the smallest diameter and multiply this by the height, and the result by 2,618. This will give the cubic inches in the barrel, and this divided by 231 will give the number of gallons.

To find the lateral pressure of water upon the side of a tank, multiply in inches, the area of the submerged side, by the pressure due to one-half the depth.

Example.—Suppose a tank to be 12 feet long and 12 feet deep. Find the pressure on the side of the tank.

144 × 144 = 20,736 square inches area of side.

12 × .43 = 5.16, pressure at bottom of tank. Pressure at the top of tank is 0. Average pressure will then be 2.6. Therefore 20,736 × 2.6 = 53,914 pounds pressure on side of tank.

To find the number of gallons in a foot of pipe of any given diameter, multiply the square of diameter of the pipe in inches, by .0408.

To find the diameter of pipe to discharge a given volume of water per minute in cubic feet, multiply the square of the quantity in cubic feet per minute by 96. This will give the diameter in inches.

To find the weight of any length of lead pipe, when the diameter and thickness of the lead are known: Multiply the square of the outer diameter in inches, by the weight of 12 cylindrical inches, then multiply the square of the inner diameter in inches by the same amount, subtracting the product of the latter from that of the former. The remainder multiplied by the length gives the desired result.

Example.—Find the weight of 1,200 feet of lead pipe, the outer diameter being ⅞ inch, and the inner diameter 9/16 inch.

The weight of 12 cylindrical inches, 1 foot long, 1 inch in diameter, is 3.8697 lbs.

⅞×⅞ = 49-64 = .765625.

9/16×9/16=81-256=.316406.

.765625 — .316406 = .449219 × 3.8697 × 1,200 = 2,086 lbs.

Set Collars.—In some cases it is possible to dispense with one set collar, due to the fact that one or more of the pulleys may come next the journal-bearing, thus taking the place of a collar at that point. Countershafts carrying loose pulleys should always have a set collar next the loose pulley opposite the tight pulley, and where a loose pulley comes near a journal box, a collar should be interposed between the box and the hub of the pulley to prevent pinching.

TABLE 36b.
FRICTION OF A COLLAR BEARING.

(As Condensed by Prof. Unwin from Mr. Tower's Experiments).

Intensity of Pressure, P, in lbs. per square inch.	Speed in Revolutions per Minute.				
	50	70	90	110	130
15	μ= .045	μ= .065	μ= .043	μ= .054	μ= .064
30	" .037	" .048	" .050	" .049	" .048
45	" .036	" .040	" .036	" .036	" .037
60	" .029	" .038	" .036	" .037	" .041
67	" .035	" .033	" .035	" .036	" .038
75	" .035	" .034	" .035	" .035	" .036
82	" .034	" .032	" .035	...	...
90	" .031	" .044	"	...	...

BELTING AND PULLEYS.

A steel tape is best in taking measurements for belting; other methods are less reliable.

A light belt on a large pulley is preferable to a thick belt on a small pulley.

The better you look after your belts the fewer machine failures you will have, which means less worry and more money.

Pulley Arms.—The arms should be well filleted at both rim and hub to render the flow of metal free and uniform in the mould. The general proportions of arms and connections to both the hub and the rim may perhaps be best developed by trial to scale on the drawing board. The base of the arm being determined, it may gradually taper to the rim, where it takes about the relation of 5/8 to 3/4 the dimensions chosen at the hub. The taper may be modified until it looks right, and then the sizes checked for strength.

Six arms are used in the great majority of pulleys. This number not only looks well, but is adapted to the standard three-jawed chucks and common clamping devices found in most shops. Elliptical arms look better than the segmental style. The flat, rectangular arm gives a very clumsy and heavy appearance, and is seldom found except on the very cheapest work.

A double set of arms may be used on an excessively wide face, but it complicates the casting to some extent.

Although a web pulley may be calculated for shear at the hub, yet it will usually be found that with a thickness of web intermediate between the thickness of the rim and that of the hub, which will satisfy the casting requirements, the requirements as to strength will be fully met.

Pulley Hub.—The hub should have a taper of 1/2 inch per foot draft, similar to that of the inside of the rim. The length of the hub is arbitrary, but should be ample to prevent rocking on the shaft. A common rule is to make it about three quarters the face width of the pulley. The diameter of the hub must be sufficient to take the wedging action of a taper key without splitting. This relation cannot well be calculated. Probably the best rule is the familiar one that, the hub should be twice the diameter of the shaft. This rule, however, cannot be literally adhered to, as it gives too small hubs for small shafts, and too large ones for large shafts. It is always best to locate the key, if possible, underneath an arm instead of between two arms, thus gaining the additional strength due to the backing of the arm. The hub is designed principally to carry the key, and through it transmit the turning moment to the shaft. Considered thus, the hub may tear along the line of the key or crush in front of the key.

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Original Gypsy Fortune Teller and Dream Book	.50	.25
The Mystic Circle Fortune Teller and Dream Book.............	.50	.25
National Policy Players' Guide and Dream Book	.50	.25
How to Tell Fortunes by Cards...	.50	.25
How to Be a Clairvoyant........	.50	.25
Zancig's New Complete Palmistry.	.50	.25

Dialect and Stage Jokes

Choice Dialect, Monologues and Stage Jokes	$0.50	$0.25
Dutch Dialect and Monologues...	.50	.25
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Negro Minstrels, Stump Speeches and Monologues	.50	.25
American Nights Entertainments.	.50	.25
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Shadow Entertainments	.50	.25
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Business and Useful Arts

Bryant's Business Guide.........	$0.50	$0.25
How to Make $500 Yearly Profit with 12 Hens.................	.50	.25
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A. B. C. Guide to Music.........	.50	.25
Photography Self-Taught........	.50	.25
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